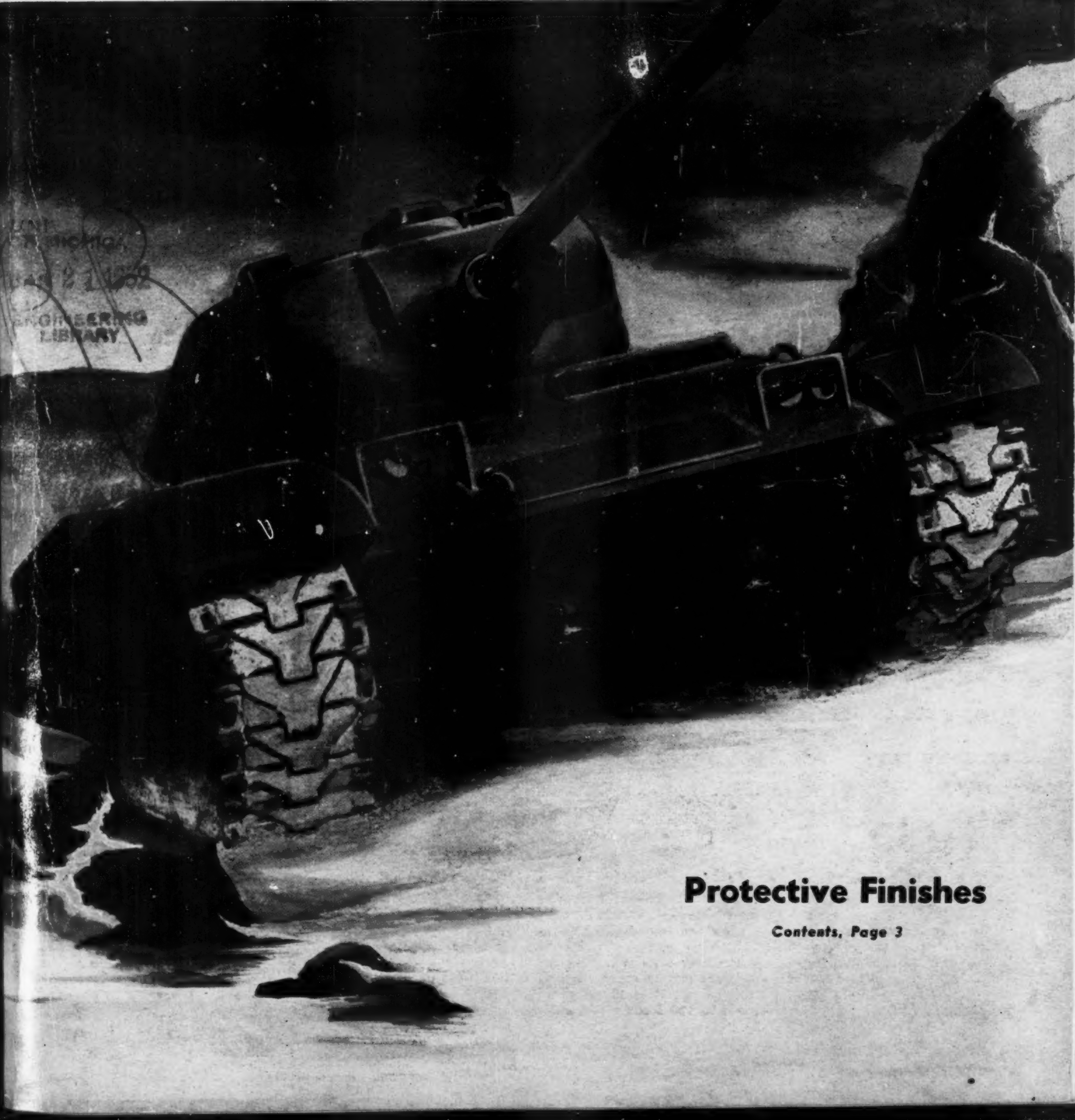
MACHINE DESIGN

January 1952



Protective Finishes

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MACHINE DESIGN

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DESIGN FOR PRODUCTION • STYLING • MATERIALS SPECIFICATION • DESIGN ANALYSIS • MACHINE COMPONENTS • ENGINEERING MANAGEMENT

Over the Board

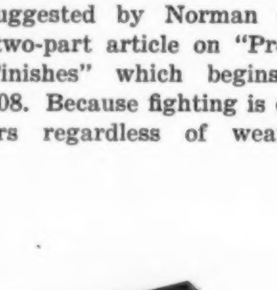
Our Expanding Staff

As part of our program to maintain MACHINE DESIGN's position as the foremost design engineering publication, we have expanded our editorial staff with the addition of Leo Spector, whose picture appears on Page 184. A native of Kansas City, Mo., Leo is a graduate of the University of Kansas (B.S. in M.E.). Prior to joining MACHINE DESIGN he was a production engineer with the Marley company where he worked on research, development and design in connection with power transmission equipment, as well as on standardization, analysis and publication of engineering data. During the war he served with the Army Air Force in the Pacific, as pilot of a P-38, and was honorably discharged in 1946 with the rank of First Lieutenant. Leo's hobbies include reading philosophy, bowling, and private flying.

This Month's Cover

Theme for this month's cover was suggested by Norman Gentieu's two-part article on "Protective Finishes" which begins on Page 108. Because fighting is done outdoors regardless of weather, adequate protection is particularly important in military equipment, as it also is in other outdoor machinery such as automobiles, trains, and farm and construction equipment.

Penton artist George Farnsworth has pictured a Walker "Bulldog" tank plowing through January snow, symbolizing the conditions under which much of the resistance to Communist aggression is being fought.



Tear Sheets and Reprints

In these columns and in correspondence we frequently refer to *reprints, tear sheets* and, sometimes, *clip sheets.* In case some of you are unfamiliar with the jargon of the publishing business, tear sheets and clip sheets are synonymous terms, and describe pages torn or clipped from the magazine. Such pages are used in filling your return-postcard requests. Reprints are separately printed individual articles without extraneous editorial material such as might appear on backing pages, and with possible addition of a title page. Because of extra "tooling-up" costs, reprints are more expensive than tear sheets, which costwise enjoy the benefit of the long production run on the entire magazine (19,000 copies).

The continuing flood of return postcards requesting tear sheets has made it necessary for us to place an expiration date on each card, approximately three months after the date on which you receive the issue. The resulting elimination of old cards will speed up our processing of new requests and give you quicker service. If you need an article after the expiration date, or if both postcards have been removed before you see the issue, we suggest you write us a letter.

Engineers as Authors

A piece under this heading in the October 1951 issue, Page 4, has elicited some interesting comments from a design engineer's secretary who prefers to remain anonymous. We had stated that in a particular organization one engineer in forty-five writes a paper or article per year. The national average of published articles is probably much less favorable and, according to our correspondent, is due to fear of competition and to professional jealousy. As she puts it, "Engineers would write articles—most of them do—but when they present them for approval the reply is 'Joe, you have written a fine article but —we cannot release it for publication, our competitors will get wise, and our patent attorney will not approve it.' Professional jealousy also has a hand in unpublished articles. Technical society subcommittees on papers and publications sometimes are quite unreasonable. Fear and jealousy do more harm in modern business than unfair competition." Strong words! Is this a fair appraisal of the situation?

MACHINE DESIGN
A PENTON PUBLICATION

TOPICS

...IN ENGINEERING AND RESEARCH

Large, Complex Parts Spray-Formed

Dense, strong metal parts of high-melting-point metals can be formed by spraying the metal onto a core, then sintering the sprayed deposit, according to a Navy Ordnance Bureau report. Large complex shapes, difficult to fabricate with powder metallurgy, precision casting or die-forging, can be built up. Strengthening of the metal is produced by the sintering process.

Rocket Sled Produces 45g Acceleration

A rocket-propelled sled, the first free-air supersonic test facility for aircraft parts, has produced accelerations up to 45*g*. Entire airplane models or component parts have been tested, and a human subject riding the sled has endured a test equivalent to stopping an airplane flying at 120 mph in a 19-foot run lasting 0.25-second. The sled, installed at the Edwards Air Force Base, travels along a precision track and is braked to a stop by a built-in scoop dipping into a trough of water 2000 feet long.

High-Temperature Coatings "Plated" on Jet-Engine Parts

High-melting-point metals, or refractory coatings of the borides, silicides, carbides or nitrides of these metals, can be deposited on jet engine parts by a new vapor "plating" process. In the process a volatile compound of the metal is passed over the work, which is heated to a temperature that will decompose or reduce the compound to form an adherent coating. Thickness can be from less than a micron (approximately 0.00004-inch) to several millimeters (about ⅛-inch).

Electronic Music Box Sounds Like Symphony

Almost any combination of musical sounds can be reproduced by an electronic music box developed by Earle L. Kent of C. G. Conn Ltd. The device, keyed by a perforated roll of paper rather than a keyboard, can play a note higher than the highest produced on any musical instrument. The operator, by manipulating the proper knobs and switches, can obtain almost any musical effect, from a solo flute to a symphony.

Iron Plating Produces Heavy Deposits

A ¼-inch buildup of practically pure electrolytic iron can be plated on worn parts by a new process developed by the Van der Horst Corp. Developed to reclaim worn and undersize cylinders, crankshafts and similar parts, the plating process uses a new bath which deposits an iron with a close-packed columnar grain perpendicular to the plated surface. Tensile strength of the deposit is approximately 75,000 psi; yield point, 60,000 psi; and Brinell hardness, 200 to 210.

Vapors Protect Metal Surfaces

Amine nitrate vapors, carried by air convection and diffusion, condense on metal surfaces to provide a thin coating which protects against rust and corrosion. The corrosion inhibitor, a powder-like substance announced by Shell Oil Co., is volatile at atmospheric temperatures, and is applied

by placing it in an airtight package or box containing the metal part, by blowing it into cylinder bores and similar enclosures, or by mixing it in a water or alcohol solution for dipping. Corrosion can be arrested at any advanced stage.

Ceramics Act Like Crystals

Ceramics formed of barium titanate may be used as crystal-type amplifiers and rectifiers in electronic circuits, according to a recent Signal Corps report, since their dielectric constants are varied by changes in applied voltage. The ceramics also have a "memory" characteristic, useful in television and computer circuits.

Welding Zinc-Alloy Die Castings

Zinc-alloy die castings are usually considered completely unweldable, but a recent report indicates that these "white-metal" alloys *can* be welded with an oxy-acetylene torch—providing necessary precautions are observed. White-metal alloys have a low melting point with a narrow range of welding temperatures, and oxide coatings readily form on the surface of the molten metal. According to a Canadian Technical Information Service report, these difficulties can be overcome by adherence to a rigid procedure, including adequate support for sections and proper welding technique.

...IN GOVERNMENT AND INDUSTRY

Military Standardization Pays Off

Instead of using as many as ten different engines for the same tank—a common World War II practice—the Army Ordnance Corps reports that new standardized designs permit four engines to be used in 11 different tanks and vehicles. Over 60 per cent of the parts in these four engines are interchangeable. A similar program is being carried out by the Corps of Engineers, which reports that 138 different small-size engines, having 15 different bore sizes and 1187 different parts, are currently manufactured in this country.

Board to Review Patent Policy

All phases of patent policy will be reviewed by a new Government board, established to recommend policies or changes in policy that would aid the national defense program. Individuals, corporations, associations and other organizations will be given an opportunity to present suggestions either orally or in written form to the three-man Patent Policy Review Board.

New Magnesium Mill To Process Large Ingots

Magnesium sheet 84 inches wide will be processed at a new hot and cold rolling mill at Madison, Wis. Ingots processed will weigh 2000 pounds, not large by steel-mill standards, but huge when compared with the 100-pound magnesium slabs usually rolled. Built by the Dow Chemical Co., the new mill will be of the tandem continuous type, rolling coils instead of single sheets.

Restricted Wartime Data Released

Reports on wartime industrial research projects, restricted until now, are rapidly being declassified and made available to industry. More than 28 million documents are scheduled for review by an Army Review Board. So far, 87 per cent of the material reviewed has been declassified. After declassification, reports are turned over to the Office of Technical Service, Department of Commerce for distribution, and many of them are listed in the *Bibliography of Technical Reports*, published by that office.

Design for Lasting Performance

GENERATIONS of design engineers have rediscovered Oliver Wendell Holmes' "Wonderful One-Hoss Shay." No engineer, the deacon who built it was probably the first man to write out a specification covering design for anticipated life. According to his philosophy, replacement of parts should be nil throughout the life of the machine, even for "a hundred years to the day."

Today's machines cannot always be so designed. Parts fatigue, creep, wear, burn, or corrode, and it would sometimes be foolish and uneconomic to attempt to build one hundred per cent life into each component. An occasional replacement at moderate cost may restore a machine to its original effectiveness with less total cost to the user throughout the life of the machine than if it had been designed like the deacon's one-hoss shay.

But there is another side to the picture. Too often complaints are heard that certain machinery manufacturers seem to be more interested in lucrative parts replacement business than in producing durable machines. However, machinery users, not accepting this situation, are turning more and more to the various hard-surfacing methods to provide a degree of wear resistance that was lacking in the new machines.

Instead of selling replacement parts the machinery manufacturers involved are placing themselves in the humiliating position of letting their customers, by rebuilding, produce better machines than the original. Existence of this state of affairs is demonstrated by the fact that wearproofing by such means as hard facing and hard inserts seems to find application more in maintenance than in original equipment.

Initiative for overcoming the situation, where it still exists, should properly come from the design engineers whose professional reputations are at stake. The methods and processes are available and well known, while the slight additional cost of adequate wearproofing can be recovered many times over. Obviously any price increase can readily be justified by the additional selling points gained. More important, in the long range, is the enhancement of the designers' and builders' reputations for producing durable machines.

Colin Carmichael

EDITOR

STRAIN GAGES AID

Positioning of Jet-Engine Blade

By E. L. Watelet
Director of Design
Precision Tools and Gages
Brown & Sharpe Mfg. Co., Providence 1, R. I.

IN THE manufacture of aviation jet engines, the exact position of the airfoil surfaces of blades, *Figs.* 1 and 2, relative to the stator or rotor on which they are mounted, is of paramount importance. Slight variations in position not only affect efficiency, but unleash unbalanced forces which become tremendous at high speeds and may cause destruction of the engine.

Usually the important relationship of the blade surfaces with respect to the mounting is obtained by machining the "root" while locating from the surfaces. Various methods have been used to accomplish this purpose, most of which have been either slow or inaccurate.

A common method which is basically sound locates the blade in a fixture from three center holes drilled relative to the blade surfaces. The resultant machined root which determines the blade position in its mounting is thus in proper relationship with the surfaces. Until recently, use of this method has been deterred by the lack of accurate and fast equipment for properly drilling the center holes. To meet this need, Brown & Sharpe Mfg. Co. developed the special-purpose machine shown in *Fig.* 3. It accurately positions jet blades, through a novel strain-gage system, and drills three center holes at the rate of 120 per hour.

During development of this machine, the basic problem involved positioning the blade so that one of its curved surfaces would be brought into coincidence with a theoretically correct surface, which in turn is in proper relation to a correct root form. The theoretical surface plane and root were established from a hardened-steel master blade. This master, complete insofar as all critical surfaces are concerned, has three center holes as shown in *Fig.* 2.

When suitably located in the machine from the root and bosses, this master serves as a means of "zeroing" sensitive strain gage elements, acted upon by probes in contact with one of the master airfoil surfaces. Once these gages are set, the problem remaining is to insert a regular blade in place of the master, to position it properly as shown by an indicating meter fed from the amplified output of the strain gages and to drill the three center holes.

Three basic elements form the machine—a cradle capable of being swivelled about two axes of rotation and movable along a vertical axis, a hinged member carrying six strain-gage actuated probes, and a set of three drill heads equipped with combination center drilling and facing cutters.

Spring Pivots Replace Trunnions

Requirements of the cradle, *Fig.* 4, involve ability to rotate the blade about the *X* and *Y* axes, *Fig.* 2, and to move in a straight line along the *Z* axis. Each motion is independent of the other two. These requirements could have been met by the use of a double trunnion arrangement on a vertical slide. However, such a design would involve bulk and inevitable looseness in the bearings, resulting in positioning errors. It was decided to eliminate both of these conditions by the use of flat spring pivots in place of trunnions.

The cradle is composed of three elements. An upper member is mounted on two flat springs, so positioned that the extensions of their planes intersect at the *X* axis. This provides rotation for a limited range about *X*. The lower ends of these springs are fastened to a second member beneath the first, which in turn is mounted on flat springs. In a similar manner they provide rotation about the *Y* axis. The lower ends of the second set of springs are fastened to a third member fastened to two horizontal parallel springs which provide vertical movement along the *Z* axis. It is to be noted that each movement is limited in range by the nature of the pivots. However, since the order of the variations in the shape of the

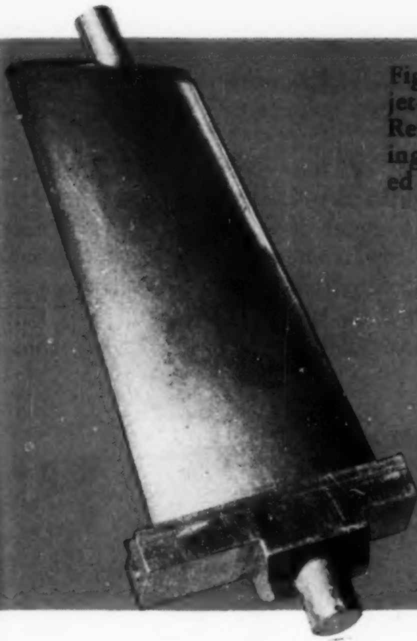


Fig. 1—Left — Unfinished jet-engine compressor blade. Reference holes for machining operations are positioned accurately with strain-gage system

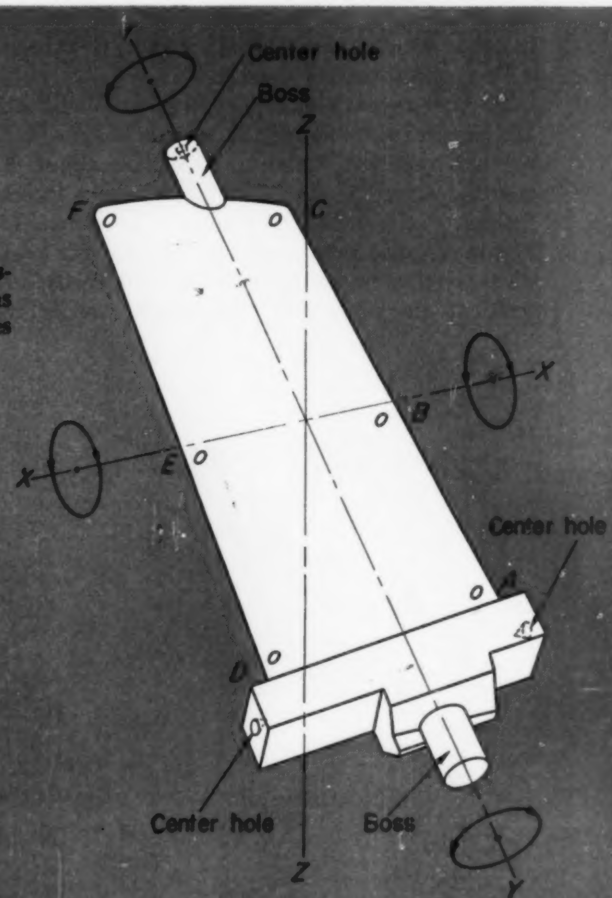


Fig. 2—Diagram of blade positioning motions and locations of the three center holes

blades is only a few thousandths of an inch, the range is entirely adequate. Each motion is positively controlled by a manually operated knob and separately clamped.

Because blades vary in size and shape, it is necessary to have individual holders for each size of blade. The blades are held in place by air-operated clamps.

In *Fig.* 5 is shown the second main element of the machine, the hinged member, having six strain-gage actuated probes or measuring points. These probes are adjustable laterally to suit blades of various sizes and are brought into contact with the blade at points *A, B, C, D, E,* and *F, Fig.* 2 by lowering the hinged member against a fixed stop.

Indicating the position of each of the six points, relative to a master blade to which the probes are set, is a simple problem. However, the requirements of correcting the position are more complex. It is necessary to show in three separate, single indications when the blade reached its proper position on the *X* and *Y* axes, and the *Z* vertical position as controlled by manipulation of the cradle.

It is obvious that separate indications from each probe would require considerable mental arithmetic on the part of the operator to position the blade, even if the blade form were identical to the master. It becomes considerably more difficult when the blade form varies from the master by having more or less twist or bow. Also, it is desired that, in the final position, the blade errors will be averaged over the six points.

Versatility of strain gages suitably interconnected solves these requirements conveniently. The basic circuit consists of six strain gages, a multiple position drum switch, and an indicating amplifier. In each position of the drum switch, the six strain gages are connected, either individually or in combinations, to give desired results on the amplifier. The first six positions, used for setup purposes give separate

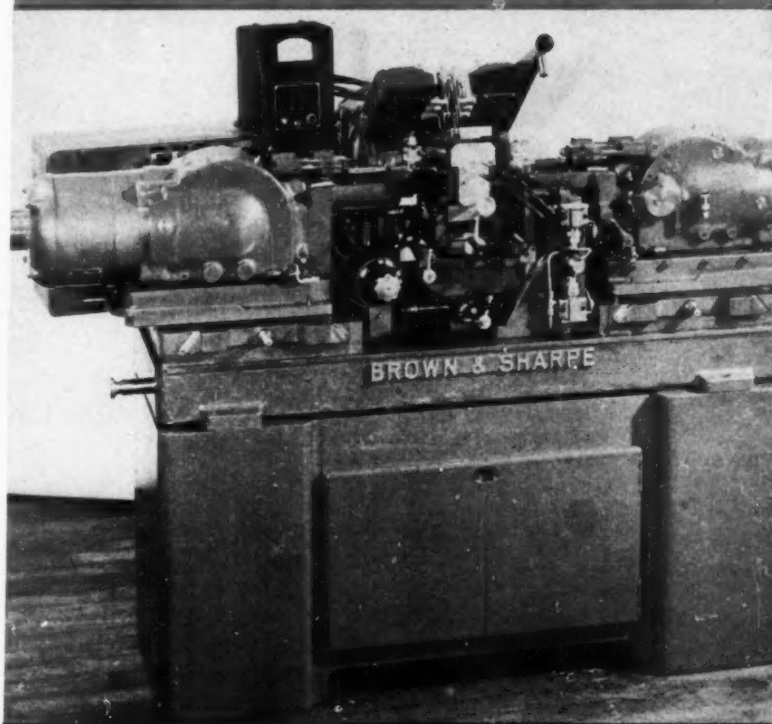


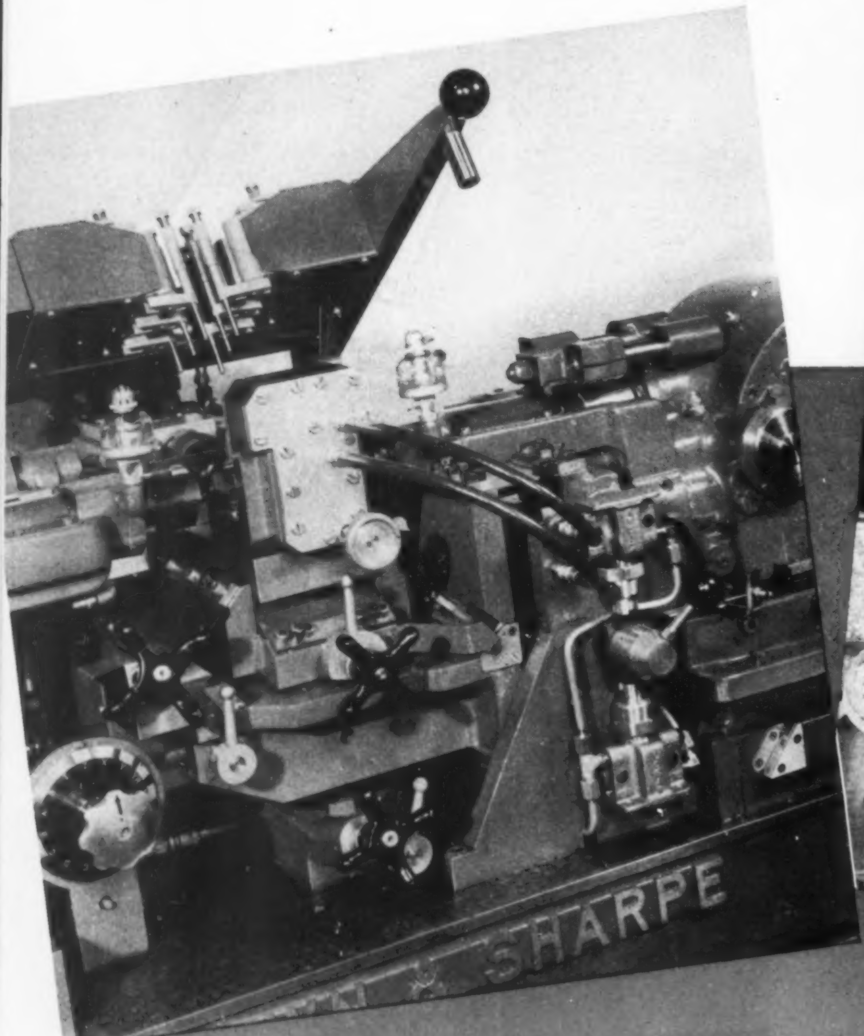
Fig. 3—Jet blade positioning and center drilling machine accurately locates three center holes for subsequent machining operations

indications at the six points on the blade, *Fig.* 2. In other words these positions serve to zero each gage with respect to the master.

In the seventh and further positions, the gages are interconnected to show as single indications, vertical position *Z*, rotary position about *Y*, rotary position about *X*, errors in twist, errors in bow, etc.

One of the measuring probes is shown in *Fig.* 6. The measuring point and associated parts are attached to a fixed member by two parallel reeds, permitting vertical movement of the point. A compensating spring, acting between the fixed member and the movable point assembly, balances the weight of the latter and is adjustable to provide the desired measuring pressure.

Vertical movement of the measuring point deflects a third reed to which are bonded four 120-ohm resistance-wire strain gages, two on each side. These four strain gages are connected to form a Wheatstone bridge circuit which acts as the sensitive measuring element. When the reed is deflected, the strain gages on the concave side are compressed while those



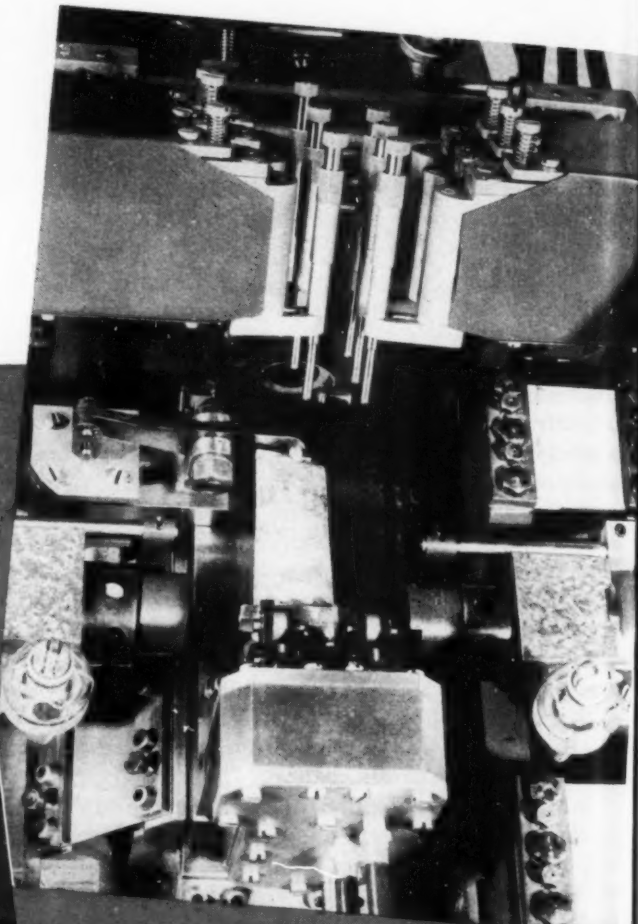


Fig. 5—Above—View of gage system showing how the six strain-gage probes contact the cradled blade

Fig. 4 — Above — Work cradle pivoted on three sets of flat springs to minimize position errors

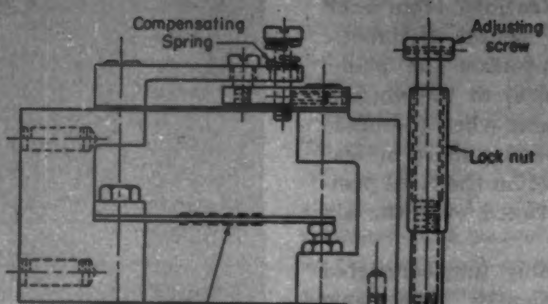


Fig. 6—Left—Measuring probe mounted on two parallel reeds to allow vertical motion of probe. Four strain gages, mounted on a third reed, indicate deflections of the measuring point

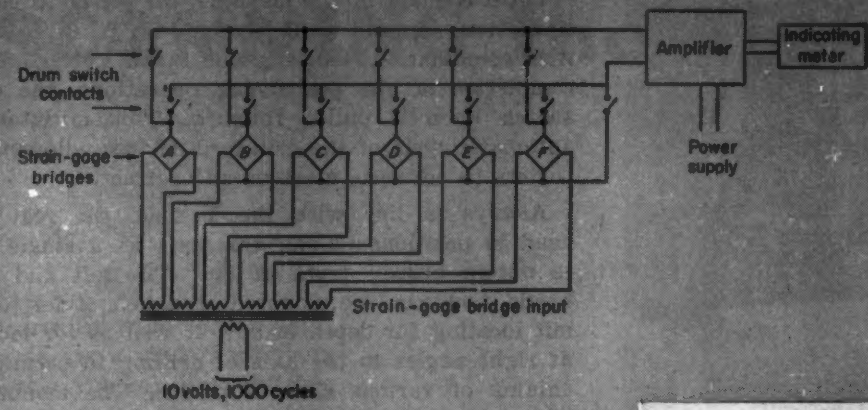


Fig. 7—Left—Schematic diagram of strain-gage bridges. Contacts are arranged in the drum switch to give combinations desired for direct readings on meter

on the convex side are elongated. The strain gages are thus all active and are so connected that their effects are additive.

Voltage output of the strain-gage bridge is in microvolts and thus requires considerable amplification to cause large indications on a meter. The output of the bridge is fed to a three-stage amplifier and then to an indicating meter. ***Fig.*** 7 shows the schematic arrangement of all six strain gage bridges. The various contacts shown are contained within the drum switch which connects the strain gage bridges to the amplifier and meter either singly or in combinations.

Indicator Shows All Positions

If any one gage, such as A, is connected to the amplifier, the meter will show the height of point A on the blade relative to that point on the master blade. If bridges A and D are connected in parallel to the amplifier, the meter will show the average height of points A and D or $(A + D)/2$. If bridges A and D are connected to the amplifier with the connections of one bridge reversed, the meter will show the average variation of points A and D about axis Y or $A - D$. Thus, by suitably interconnecting the bridges, any combination of average variations may be determined. By properly moving the cradle, the blade to be centered can be located in the best position to coincide with the master setting. If errors in twist or bow are present, these errors can be averaged over the surface of the blade.

The following example will illustrate interconnection of the gages for a specific position of the switch. With the switch at position Y, it is desired to manipulate the cradle to rotate the blade about the Y axis until its X is parallel with the X axis of the master blade. Electrically subtracting the average of the arithmetical sum of the gage outputs at A, B, and C from that of D, E, and F, the meter indication shows the deviation of the X axis from *the desired position.* The cradle is then rotated about Y, until the meter indicates zero. In a similar manner the blade is positioned about the X axis and along the Z axis until all axes coincide. Thus, the blade is properly positioned in three simple operations by observing three separate readings on an indicating instrument.

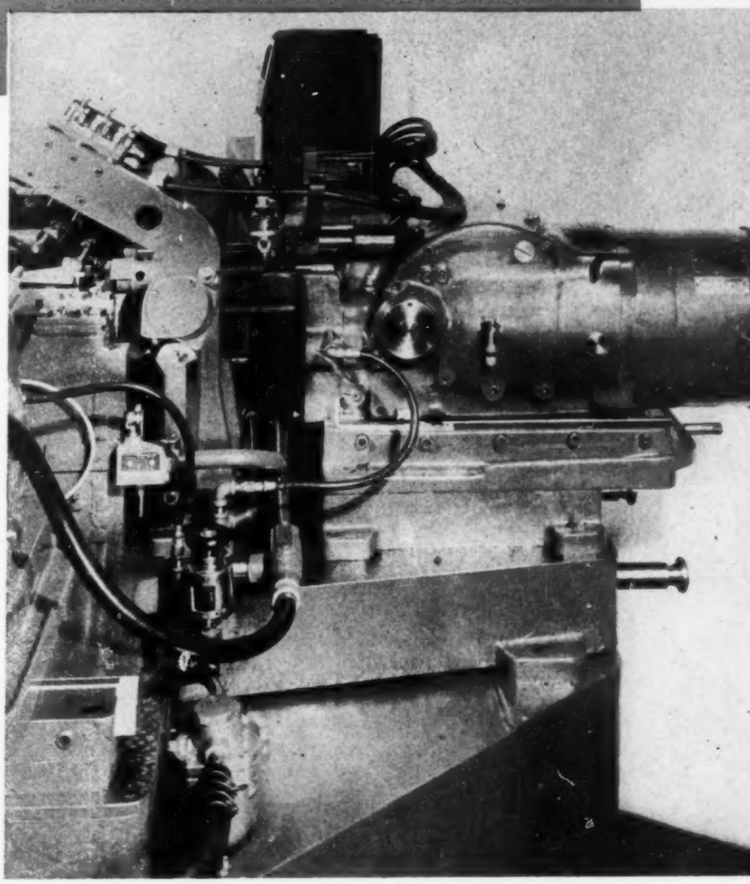


Fig. 8—Rear drill head mounted on a single slide. It center bores and faces automatically when the blade is positioned by zero readings on indicating instrument

Optional additional positions of the drum switch provides information such as errors in twist or bow to determine whether or not it is acceptable for use. A particular advantage of strain-gage position indication is the fact that small errors in the airfoil curved plane of the blade are averaged, so that in the zero position the errors will not all show up at one corner or one edge of the blade. The operator need pay no atten-

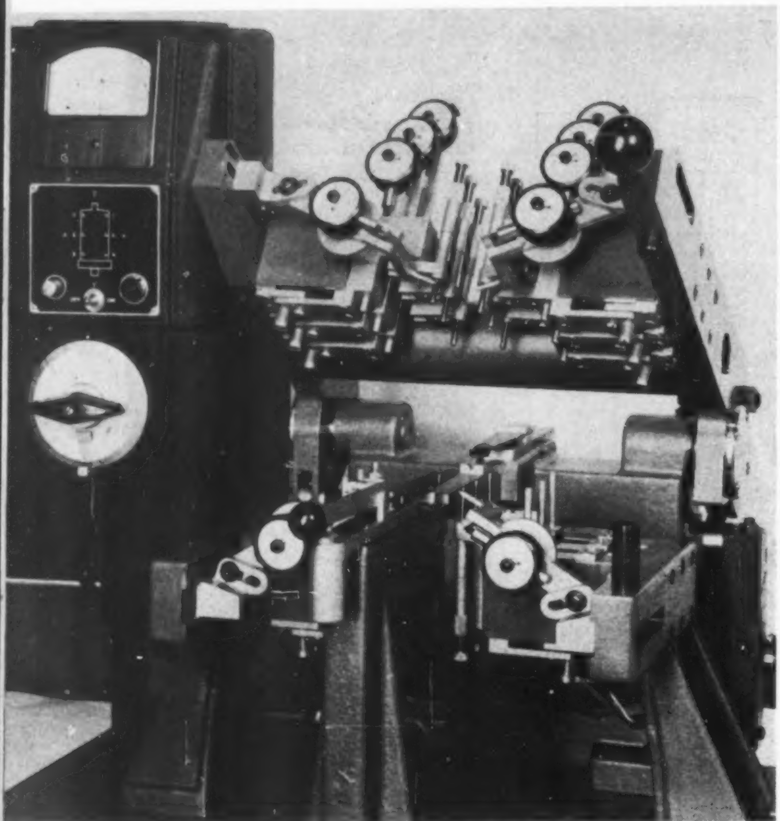


Fig. 9—Blade inspection gage for inspecting precision forged blades before further processing

tion to these errors, since the machine takes care of them automatically.

Third element of the machine, the set of three motor-driven cam-fed drill heads, *Fig.* 8, is equipped with combination centering and facing cutters. Upon completion of the positioning operations, the drum switch knob is pulled forward. This initiates the automatic cycle of the drill-feed devices, allowing the cutters to advance, machine and retract.

Always in line with the *Y* axis, the rear drill head is positioned along this axis on a single slide to obtain proper depth of cut. The left and right heads, however, are mounted on double slides to permit locating for depth of cut, as well as for location at right angles to the axis of drilling to permit machining of various sizes of blades. The machine is adjustable to position and center all sizes of blades used in the compressor unit of a specific jet engine.

This strain gage system has also been used for the purpose of inspecting the airfoil surfaces of precision forged blades. The gage shown in *Fig.* 9 was developed to check the contour and thickness of blades before they are processed in the positioning and centering machine.

Because the jet-engine industry is relatively new, it continually meets unusual production problems. Devices such as those discussed help solve some of these problems and will aid the industry to economically supply the nation's needs for its products.

Torque Converter Acts as Brake

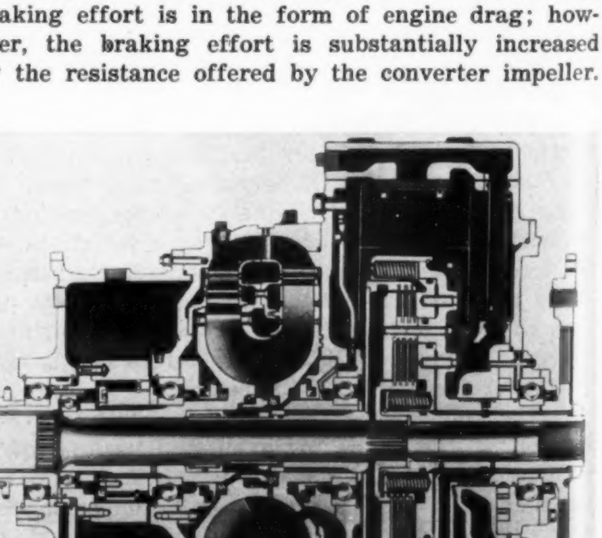
A THREE-STAGE hydraulic torque-converter transmission for trucks eliminates 99 per cent of forward gear shifting on tough grades and performs 90 per cent of the braking. The new unit, designed by Twin Disc Clutch Co., has been field tested under grueling hauling requirements on the Mesabi Iron Range.

The five-speed transmission provides a reverse gear, a power take-off drive for the truck dump body and low gear ratio for unusually bad holes. Fourth speed of the transmission is a direct drive, and it is in this ratio that the torque converter does 99 per cent of its payload upgrade hauling on the Mesabi range, even though grades vary on an average from 8 to 13 per cent. Loads up to 30 tons on grades up to 13 per cent have been handled.

Designed to exert braking effort in excess of engine rated horsepower, the automatic torque converter brake can perform 90 per cent of the braking job in returning to the bottom of the pit. It reduces air demand on the engine air compressor 50 per cent and prolongs wheel-brake life.

The operator's control valve is equipped with four selections: neutral, torque-converter drive, mechanical drive, and combined engine and hydraulic braking.

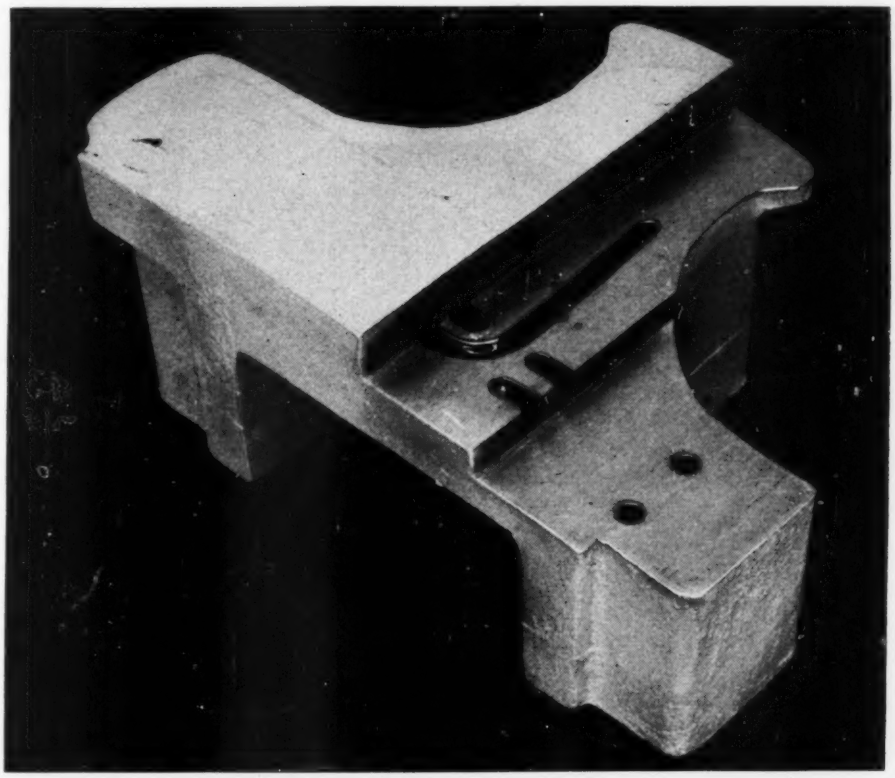
Hydraulic braking is provided by movement of the control valve to the braking position, normally when the converter is operating in direct drive. In this position, the control valve spool uncovers both the direct mechanical drive clutch ports and the hydraulic drive clutch ports, thus driving the converter, direct drive shaft and engine at a common speed, resulting in maximum braking effort. Part of this braking effort is in the form of engine drag; however, the braking effort is substantially increased by the resistance offered by the converter impeller.

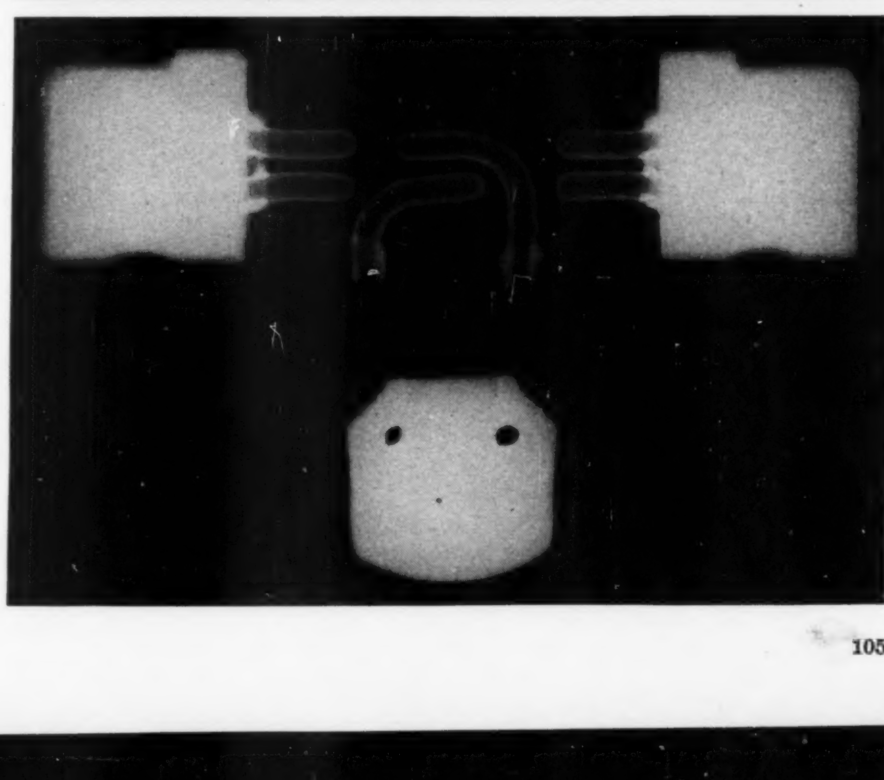


SCANNING the Field For Ideas

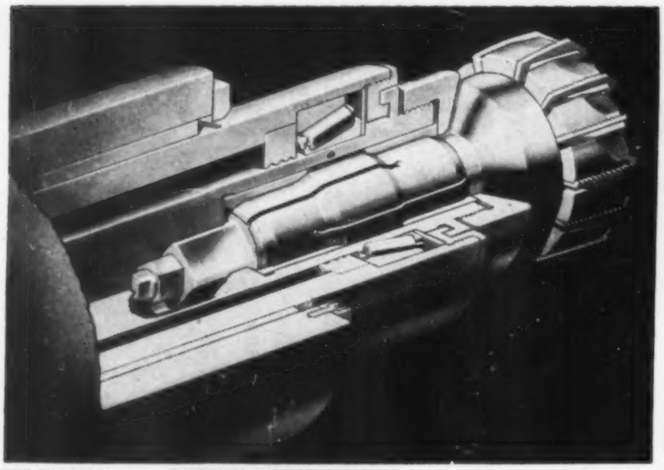
Hydraulic manifold, right, connects servo units with their actuating controls by utilizing stainless steel tubing cast into an aluminum housing. The tubing, bonded in the casting by a process developed by the Al-Fin Division of Fairchild Engine and Airplane Corp., is shaped to form passageways through the housing to connect the cylinders and valves. This simple and lightweight housing contains not only the passages but also the actuating pistons. As shown in the sectioned and X-ray views the hydraulic lines are formed to follow the most direct paths possible. The three projecting portions of the casting are bored to form hydraulic cylinders.

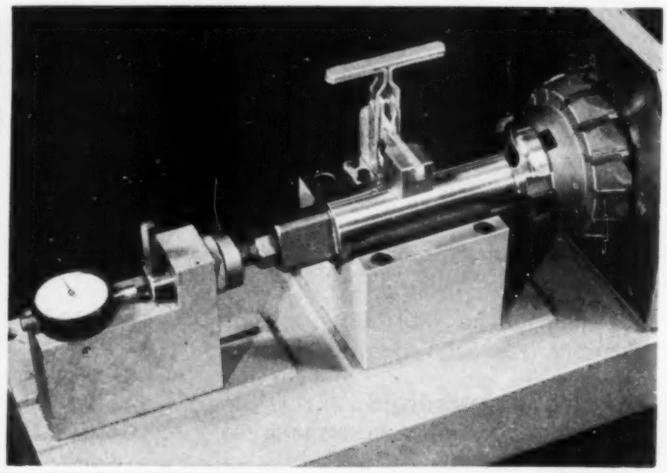
The tubing sections have blind ends, fluid being admitted to the cylinders through slots milled into the tubing. Drilled holes tap the lines for connection to the control valve body. Sealing at the control section joint is accomplished by O-rings. Besides being compact and light, the unit simplifies installation considerably because numerous high - pressure hydraulic connections are eliminated by a single, easily mounted housing.

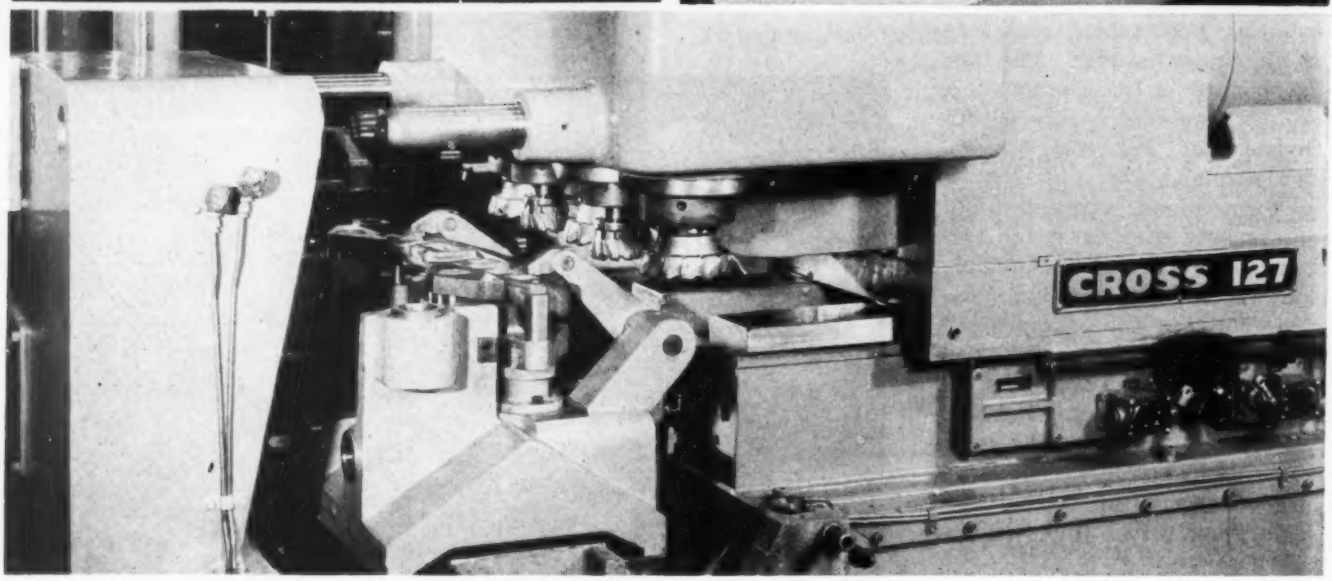




Preset tools in multiple-spindle milling heads reduce machine down time by facilitating tool changing and by providing accurate tool positioning. The spindle drive, developed by The Cross Co. and employed with these tools, is shown in a milling machine and in a sec-

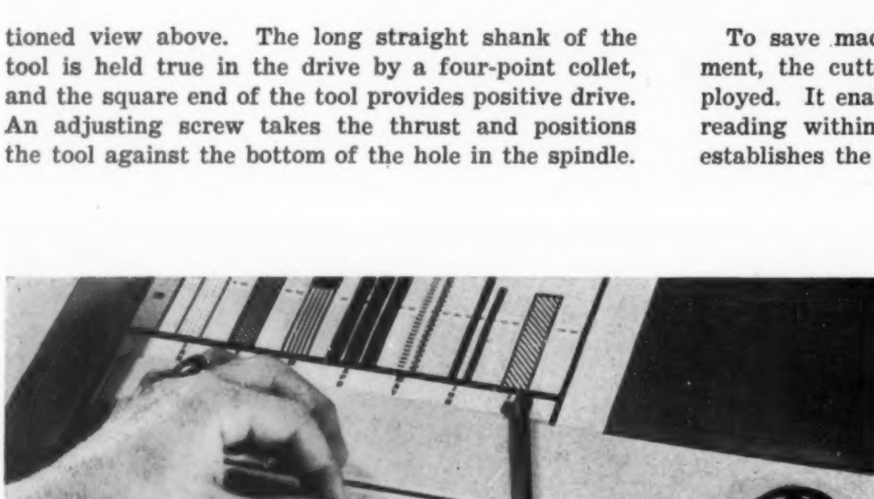






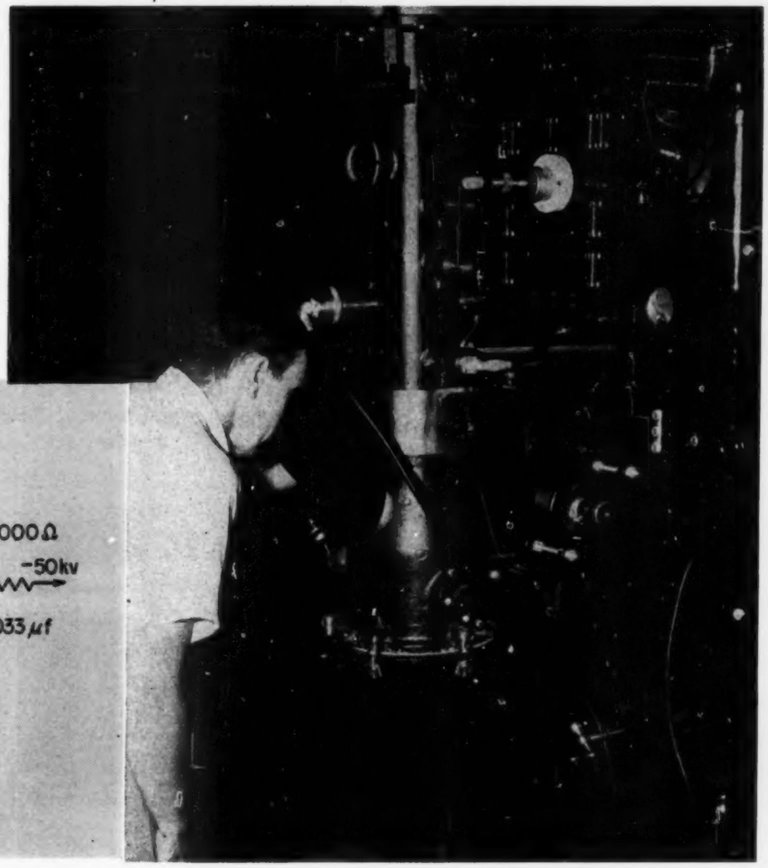
tioned view above. The long straight shank of the tool is held true in the drive by a four-point collet, and the square end of the tool provides positive drive. An adjusting screw takes the thrust and positions the tool against the bottom of the hole in the spindle.

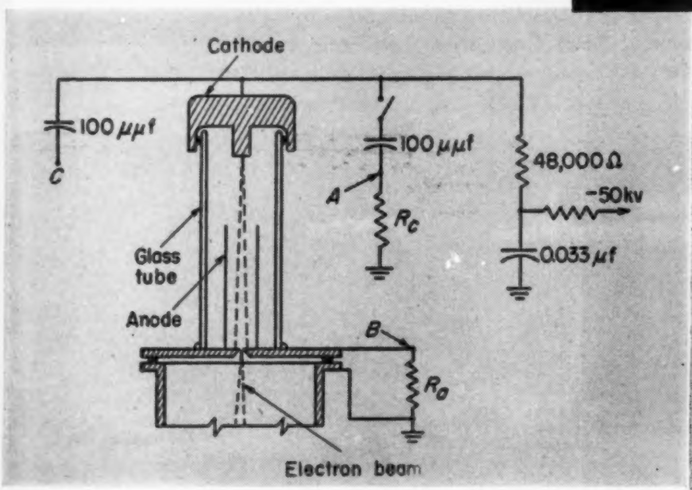
To save machine time and simplify cutter adjustment, the cutter setting fixture shown above is employed. It enables setting the tool to a dial indicator reading within 0.0005-inch. A master length gage establishes the limits on the indicator.



Time-saving method of producing statistical bar charts, left, utilizes printed tapes which adhere to Vinylite plastic sheet. Grid lines, printed on the sheet in nonphotographic blue, serve as guides in aligning the adhesive-backed tapes bearing printed months, figures, lines, or bars of various widths, patterns and forms. When reproduced by photographic methods, the master chart can be brought up to date readily or the tapes completely removed. The system, developed by Chart-Pak Inc., also includes blocks for organizational charts.

Ideas

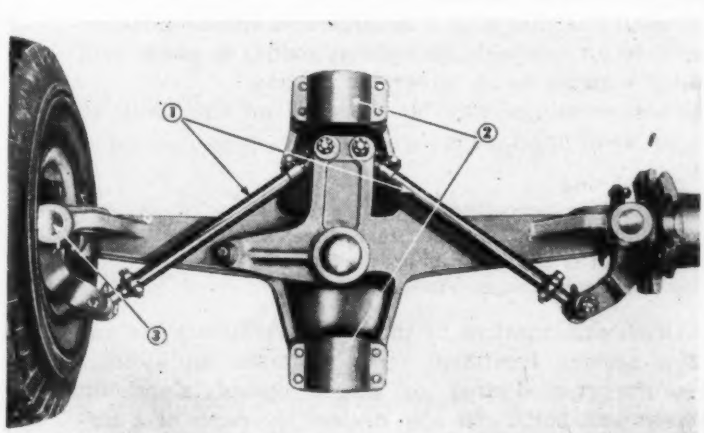




High-speed writing oscillograph, above, makes clearly visible traces at three-fourths the velocity of light for studying rapidly varying electrical surges. In the method, developed by the National Bureau of Standards, a momentary pulse having a peak value of about 2000 volts is superimposed on the steady d-c voltage of 50 kv normally applied to the discharge tube of a cathode-ray oscillograph. The pulse duration is about 5 microseconds, producing a sudden increase in the voltage across the tube within 0.5-microsecond. This sudden change disrupts equilibrium conditions in the tube, producing a greatly intensified beam.

Illustrated above are schematic circuits, showing two alternative methods for increasing the intensity of the electron beam. In the first method the switch above A is left open. When a current pulse is passed through R_a, the potential difference between the anode and cathode is given a similar pulse which increases the beam intensity. For the second method the switch A is closed, connecting the cathode to one terminal of a small high-voltage capacitor. The other terminal of the capacitor is connected to ground through resistor R_c. The capacitor is charged to 50 kv by the steady voltage on the cathode. When a current pulse is passed through R_c, the cathode voltage is changed by the drop through R_c. Duration of this change in voltage is determined by the small capacitance in series with the 48,000-ohm resistor, giving the synchronized time duration of the intensified trace.

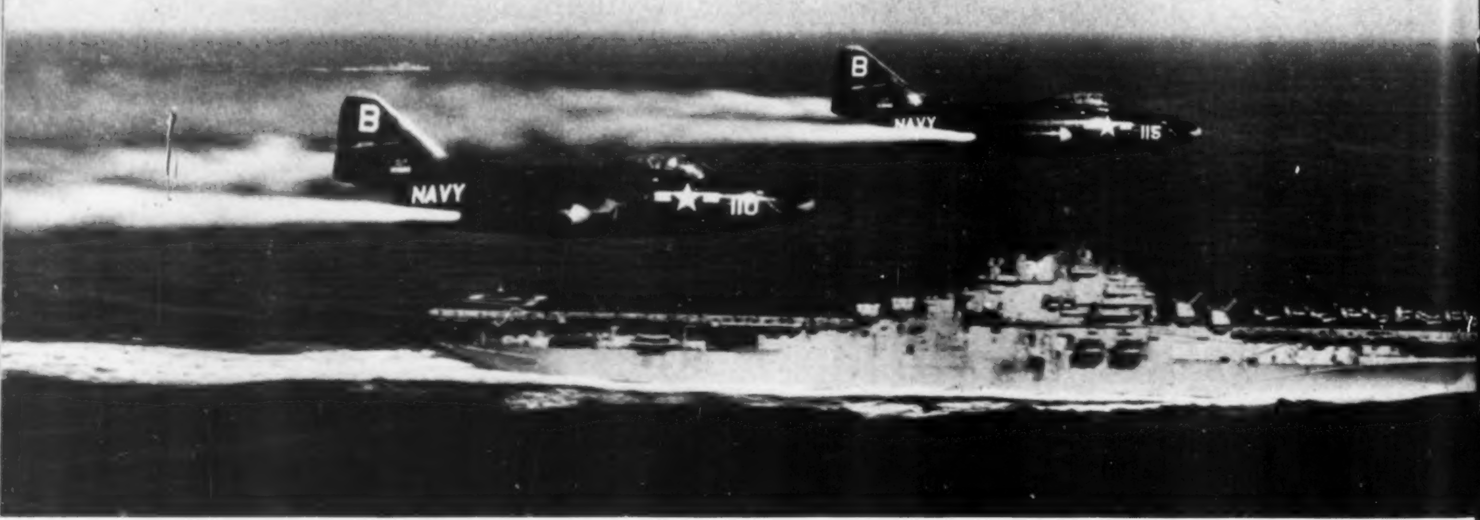
Improved steering right, for industrial trucks is obtained through a new tierod arrangement and torsional rubber bushings. Employed on Clark Equipment Co. trucks, the tierods (1) are more nearly in line with the forces they transmit. Eliminating conventional springs, torsional rubber bushings (2) are used at both pivot mount points to absorb shocks that otherwise would be transmitted to the frame, resulting in easier riding and steering qualities. Also, this design brings the center of the wheel closer to the king pin (3), and reduces road shock. Also, due to the arrangement of the members, road clearance is increased considerably.



PROTECTIVE FINISHES

. . . phosphate coatings for military equipment

By Norman P. Gentieu
American Chemical Paint Co.
Ambler, Pa.



Part 1—Finishes and Undercoating

CHEMICAL conversion coatings may be defined as inorganic films formed on metals by chemical reaction of the metal surface with a suitable medium. The films may be oxide, sulfide, chromate, phosphate, or chromic phosphate. Regardless of the quality of the end product or the objective of the end result, the principle of conversion coatings always remains the same—the surface of a metal is changed by chemical reaction with a natural or artificial environment to an insoluble phosphate, oxide, or other salt which remains as an integrated coating.

These coatings may be classified on the basis of objective of application:

1. Coloring
2. Rust or corrosionproofing
3. Improvement of wear resistance, safe "breaking-in" of friction or rubbing surfaces
4. Improving paint adhesion.

Often combinations of these factors dictate the use of a specific treatment. For example, an aircraft manufacturer desired to corrosionproof aluminum screws and bolts. He also desired the parts of a certain size to be easily distinguishable from similar parts differing slightly in size. He found that by anodizing (electrolytic oxidizing) the parts in sulphuric acid and dyeing them in variously colored dyes he was able to achieve both objectives.

For strategic reasons during warfare it is desired to make firearms as inconspicuous as possible. This is particularly true of small arms which move with troops and which, if shiny, may give positions away to the enemy. It is also important for functional reasons to prevent corrosion. During World War I and prior to that time small arms were blackened by chemical treatments such as Carbonia Black, Nitre Blueing or the Browning Treatment. All of these were similar in that they resulted in a black ferroso-ferric oxide coating which had to be kept oiled. All of them were of limited value because they lacked the corrosion-resistance necessary for adverse field conditions and because the coating burnished to a reflecting surface.

The emergence of the phosphate treatment of steel, stemming from Coslettizing invented by Coslett in England about 1898, changed the entire picture of finishes for military arms. During World War II most of our small arms were finished with a proprietary zinc-base phosphate finish. More recently a change to the manganese-phosphate base bath has

been made because it was found that this type of finish holds up better under the abuse of field handling. It consists of a microcrystalline film of manganese and iron phosphates impregnated with chromic acid and a rust preventive oil or cutback petrolatum. On rifles and machine guns it has shown excellent wear and corrosion resistance and has a dull black nonreflecting surface.

Black to gray-black corrosion resisting finishing can be applied to steel by phosphatizing processes. These yield crystalline, insoluble, nonconducting zinc-iron or manganese-iron phosphate films with coating weights from 1000 to 3000 mg per sq ft. They are usually given a final aqueous rinse in a weak chromic acid bath which greatly enhances their corrosion resistance. Finally, they are "sealed" with a rust preventive oil or compound or wax. The nature of these coatings is such that they adsorb and hold large quantities of the sealer yet present a surface "dry" to the touch. Certain of these coatings, based on manganese phosphate are excellent bases for paint and also serve as a nongalling finish for friction surfaces. This will be discussed in greater detail.

Two Panther jets flying over the USS Princeton. All painted aluminum surfaces of the Panthers are pretreated with chemical film. Steel parts are treated with zinc phosphate and painted to give a type II Class B finish. Friction surfaces in the aircraft engine are treated with phosphate rustproofing

Artillerymen firing a 105-mm howitzer. Gun mount and recoil mechanism are given a phosphate rustproofing. The projectiles receive a Grade I finish of zinc phosphate and paint

Chemical conversion coatings are more important to the National Military Establishment today than ever before. During World War II, the extreme urgency of filling up the logistic pipe lines from our factories to the fighting front resulted in neglecting suitable chemical pretreatment and painting of many items. This was true of shells, bombs, rockets, cartridge cases, ammunition boxes, rocket and shell tanks, blitz cans, 55-gallon drums, aircraft, equipage hardware. Today, or in the near future, these items—representing billions of square feet of surface of steel, zinc and aluminum—will require Grade I, JAN-C-490 for steel, MIL-C-5541 for aluminum and equivalent specifications for other metals depending on individual requirements.

Phosphate Coatings

Orthophosphoric acid and its derivatives play a stellar role in the drama of phosphate coating protection. This is so because their primary phosphate salts of the bivalent metals—iron, zinc, and manganese—are water soluble and provide chemical solutions in which metal parts can be coated; their secondary salts, excepting zinc, are sparingly soluble; and their tertiary salts are highly insoluble and stable. It is primarily the tertiary salts which appear in phosphate coatings and form an important substrate finish by virtue of their insolubility. Other important properties of phosphate coatings are

1. Electrical insulating value which tends to suppress galvanic corrosion under the paint film, tending to restrict corrosion to damaged areas
2. Physical form of the crystalline finish, tending to "key" the organic finish film in the interstices of the phosphate crystals
3. Adhesion to the basis metal with which the film is integrally bound.

The following equations, referable to a zinc-phosphate process, indicate the complex mechanism of phosphate finish formation:

1. Pickling or Corrosion Reaction

$$Fe + 2H_3PO_4 = Fe(H_2PO_4) + H_2\uparrow \quad (1)$$

2. Hydrolysis Reactions of $Zn\ (H_2PO_4)_2$:

$$Zn(H_2PO_4)_2 \overset{H_2O}{\rightleftharpoons} ZnHPO_4 + H_3PO_4 \quad (2)$$

$$3ZnHPO_4 \overset{H_2O}{\rightleftharpoons} Zn_3(PO_4)_2\downarrow + H_3PO_4 \quad (3)$$

$$3Zn(H_2PO_4)_2 \overset{H_2O}{\rightleftharpoons} Zn_3(PO_4)_2\downarrow + 4H_3PO_4 \quad (4)$$

3. Oxidation of Ferrous Phosphate:

Chance Vought Cutlass, a carrier-based twin-jet capable of flying faster than any military jet craft in production, has all aluminum parts treated with a chemical film followed by a zinc chromate primer and finished with cellulose nitrate lacquer

A. By atmospheric Oxygen:

$$4Fe(H_2PO_4)_2 + O_2 = 4FePO_4\downarrow + 2H_2O + 4H_3PO_4 \quad (5)$$

B. By Peroxide (as in Cold Spray Grandine No. 30)

$$2Fe(H_2PO_4)_2 + H_2O_2 = 2FePO_4 + 2H_2O + 2H_3PO_4 \quad (6)$$

C. Nitrite (as in Spray Granodine *M*)

$$Fe(H_2PO_4)_2 + NaNO_2 = NaH_2PO_4 + NO\uparrow + FePO_4 + H_2O \quad (7)$$

Reaction in Equation 1 shows the prime mover of the phosphatizing reaction, the oxidation of the iron surface by hydrogen ion or uncombined acid. If the amount of this acid in relationship to the zinc phosphate is too great, only etching or pickling of the surface will result, owing to the failure to supersaturate the solution adjacent to the work with tertiary zinc phosphate. If the amount of uncombined acid is too small, on the other hand, either no oxidation of the steel will occur or it will proceed so slowly as to be impractical. Also, if the amount of uncombined acid is too small, normal hydrolysis of the bath will proceed rapidly, causing a precipitation of zinc phosphate as a sludge (Equations 3 and 4).

Thus a properly balanced coating bath must contain a suitable ratio of uncombined acid to primary acid phosphate salt to yield the proper oxidation rate of the iron and also permit rapid supersaturation of the solution layer adjacent to the work.

The function of oxidizing agents in phosphatizing baths of the type under discussion has been interpreted variously. For example, hydrogen peroxide and sodium nitrite serve, as shown in Equations 6 and 7,

Table 1—Military Finish Specification Data

Metal	Government Specification	Specification Title	Typical Chemicals
	JAN-C-490, Grade I	Cleaning and Preparation of Ferrous Metal Surfaces for Organic Protective Coatings	
	U.S.A. 57-0-2 Type II, Class C	Finishes, Protective, for Iron and Steel Parts	
	U.S.A. 51-70-1, Finish 22.02, Class C	Painting and Finishing of Fire Control Instruments; General Specification for	
	MIL-E-917A (Ships)	Equipment, Electric Power, Basic Requirements for (Naval Shipboard Use)	
	MIL-S-5002	Surface Treatments (except Priming and Painting) for Metal and Metal Parts in Aircraft	
	MIL-V-3329	Vehicles, Combat, Self-Propelled and Towed; General Requirements for	Granodine Bonderite #100
	JAN-F-495	Finishes for Equipment Hardware	
	JAN-T-704	Treatment and Painting (for Construction and Engineering Equipment)	
	AN-F-20	Finishes, for Electronic Equipment	
	U.S.A. 50-60-1	Containers, Metal, for Artillery and Rocket Ammunition	
	16E4 (Ships)	Electronic Equipment, Naval Ship and Shore: General Specification	
Steel	U.S.A. 57-0-2, Type II, Class B	Finishes, Protective, for Iron and Steel Parts	
	U.S.A. 51-70-1, Finish 22.02 Class B	Painting and Finishing of Fire Control Instruments; General Specification for	
	MIL-C-16232, Type II	Coatings—Phosphate; Oiled, Slushed, or Waxed (for Ferrous Metal Surfaces) and Phosphate Treating Compounds	Permadine Parco Compound
	JAN-L-548A	Link, Metallic Belt, 20 mm., M8	
	AN-F-20	Finishes, for Electronic Equipment	
	M-364	Navy Aeronautical Process Specification for Compound Phosphate Rust-Proofing Process	
	U.S.A. 57-0-2, Type II, Class A	Finishes, Protective, for Iron and Steel Parts	
	U.S.A. 51-70-1, Finish 22.02, Class A	Painting and Finishing of Fire Control Instruments; General Specification for	
	MIL-C-16232	Coatings—Phosphate; Oiled, Slushed, or Waxed (for Ferrous Metal Surfaces) and Phosphate Treating Compounds	Thermoil Granodine Parco Lubrite
	AN-F-20	Finishes, for Electronic Equipment	
	M-364	Navy Aeronautical Process Specification for Compound Phosphate Rust-Proofing Process	
	MIL-C-5541	Chemical Films for Aluminum and Aluminum Alloys	
	MIL-S-5002	Surface Treatments (Except Priming and Painting) for Metal and Metal Parts in Aircraft	
Aluminum	AN-F-20	Finishes, for Electronic Equipment	Alodine Iridite Al-Coat #14
	U.S. Navord O.S. 675	Specifications for the Manufacture and Inspection of Cartridge, Powder, and Rocket Tanks (Aluminum)	
	16E4 Ships	Electronic Equipment, Naval Ship and Shore: General Specification	
	QQ-P-416	Plating, Cadmium (Electrodeposited)	
	JAN-F-495	Finishes for Equipment Hardware	
	AN-F-20	Finishes, for Electronic Equipment	Lithoform Bonderite #32 or #34
Zinc	U.S.N. Appendix 6	Instructions for Painting General Specifications for Building Vessels of the United States Navy	
	MIL-E-917A (Ships)	Equipment, Electric Power, Basic Requirements for (Naval Shipboard Use)	

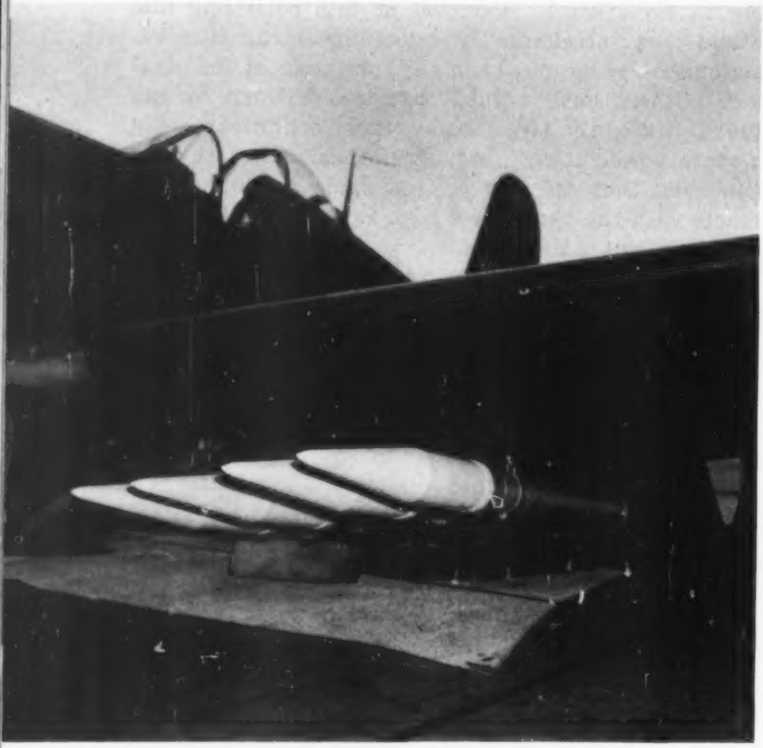
to keep the baths free from ferrous phosphate, a constituent which, if permitted to build up, decelerates the action of the baths. Peroxide oxidizes each molecule of ferrous phosphate with the liberation of a molecule of free phosphoric acid (Equation 6). To compensate for this release of acid which would otherwise throw the bath out of balance, a solution of sodium hydroxide is added in metered quantities to certain types of phosphating baths to keep the pH or free acidity on an even keel.

In other processes, on the other hand, sodium nitrite purges the bath of ferrous phosphate without the objectionable release of free acid (Equation 7). In the view of certain investigators another function of oxidants in these baths is as depolarizers, to remove a "hydrogen blanket" which would form according to Equation 1. Removal of this polarizing film would then "accelerate" the pickling or corrosion reaction, the prime mover in the formation of the coating. Others have argued that the oxidants oxidize the iron directly (iron, under these conditions being a more powerful reducing agent than nascent hydrogen) and that the iron or iron oxide simultaneously reacts with the phosphoric acid. Regardless of theory, the end result is the same.

For coating to occur, that portion of solution in contact with the steel must be supersaturated with respect to the coating phosphates. This local supersaturation is caused by the steel entering into solution due to its oxidation by hydrogen ion (free acid) or

Table 2—Protective Finish and Corrosion Resistance Data

Government Finish Specification	Metal Treated	Typical Products Treated	Type of Coating	Final Finish	Salt-spray Resistance Required
JAN-C-490, Grade I	Steel Iron	Iron and steel components other than mechanical, automotive, or other equipment or material for which there are applicable specifications	Zinc phosphate	Paint: cellulose lacquers; synthetic enamels, etc.	250 hours (when coated with 0.7 to 1.2 mils of enamel or zinc chromate primer)
U.S.A. 57-O-2C Type II, Class C	Steel Iron	Iron and steel parts of U. S. Army materiel	Zinc phosphate	Paint	250 hours (after painting). It is preferable to specify the minimum number of hours of salt-spray test for the particular paint system required, since different paint systems will differ in their salt-spray resistance.
U.S.A. 51-70-1A Finish 22.02, Class C	Steel Iron	Fire-control instruments	Zinc phosphate	Various organic finish systems as given in Specification	250 hours
MIL-E-917A (Ships)	Steel Iron	Electric (but not electronic or interior communication) power equipment for Naval shipboard applications	Zinc phosphate	Two coats of gray enamel, conforming to Type II or III, Class 2 of Specification MIL-E-15090, shall be applied as continuous films, each approximately 0.001-inch thick	250 hours
MIL-V-3329	Steel Iron	Combat vehicles, self-propelled and towed	Zinc phosphate	Olive-drab semigloss enamel for exterior surfaces	250 hours
U.S.A. 50-60-1D	Steel Iron	Metal containers for artillery and rocket ammunition	Zinc phosphate	Olive-drab enamel	250 hours
MIL-C-5541	Aluminum	Military aircraft	Chemical film	Zinc chromate primer; and cellulose nitrate lacquer	500 hours
MIL-S-5002	Aluminum	Metal and metals parts in aircraft	Chemical film	Paint	500 hours
U.S. Navord O.S 675	Aluminum	Cartridge powder and rocket tanks	Chemical film	Blue-grey enamel, Bureau of Ordnance Specification 52 E 13 (Ord)	150 hours
16E4 (Ships)	Aluminum	Naval ship and shore electronic equipment	Chemical film	As specified in the individual equipment specification	192 hours for frame and enclosure structures and 48 hours for individual parts.
QQ-P-416	Cadmium-plated steel	Various items except objects employed for use as a food container or cooking utensil	Phosphate	Paint	
JAN-F-495	Zinc and zinc-plated steel	Equipment hardware	Zinc phosphate	Enamel	200 hours
MIL-E-917A	Zinc	Electric (but not electronic or interior communication) power equipment for Naval shipboard application	Zinc phosphate	Paint	
MIL-C-16232 Type I	Steel Iron	Friction or rubbing surfaces such as pistons, piston rings, gears, cylinder liners, camshafts, rocker arms, etc.	Manganese-iron phosphate	Rust-inhibiting lubricating oil	without finish—1½ hours with finish—24 hours
U.S.A. 57-O-2C, Type II, Class A	Steel Iron				without finish—1 hour with finish—24 hours
U.S.A. 51-70-1A, Finish 22.02, Class A	Steel Iron				without finish—1 hour with finish—24 hours
MIL-C-16232 Type II	Steel Iron	Nonmoving parts—nuts, bolts, screws, hardware, tools, guns, cartridge clips	Zinc phosphate	Rust preventive compound	without finish—2 hours with finish—48 hours
U.S.A. 57-O-2C, Type II, Class B	Steel Iron				without finish—2 hours with finish—36 hours
U.S.A. 51-70-1A, Finish 22.02, Class B	Steel Iron				without finish—2 hours with finish—36 hours



Corsair carrying new antitank rockets which have a Grade I finish of zinc-phosphate coating and paint over entire surface including head, body and motor

other oxidizing agent. The nearer the solution as a whole approaches a high state of supersaturation, the less iron is required to enter the solution for coating to occur, and vice versa. When this local supersaturation is attained, zinc phosphate crystallizes out of solution on the steel, forming the desired coating. This crystallization occurs on a nucleus of, perhaps, a phantom coating of iron phosphate which cannot all dissolve in the supersaturated layer. Iron phosphate of 15-25 per cent will remain in the resultant coating.

Practical treatments have been developed for protecting various base metals. Among these are the Government specifications for finishes on military equipment, shown in TABLE 1, indicating treatments for steel, aluminum and zinc. In TABLE 2, the metal treated and typical products are listed with respect to coating, finish and resistance required. These specifications will be discussed for pretreating steel, aluminum and zinc; rustproofing; and protecting friction surfaces.

Improving Paint Adhesion on Steel

Originally, phosphate coatings were utilized only when severest exposure conditions were encountered; but now, with processes adapted to small as well as to large production, durability and lasting beauty are imparted by this protective treatment to countless products in all industries.

On metal surfaces—usually, steel, iron, zinc and aluminum—that have been freed from rust, oil and soil, phosphate coatings are formed by dipping the work into the coating baths, carrying it through the several sprays in a multistage power washer, or applying the coating solutions with a brush. The method to be used is determined by the type of work, its size and the number of pieces to be produced.

In all cases, the final rinsing of the coated work preparatory to painting is an important part of the process, for salts in untreated tap water leave objectionable paint blistering and shedding residues. These are destroyed by the addition of the proper amounts of chromic or chromic and phosphoric acids to the final rinse bath.

Ordnance materiel for which a zinc phosphate coating is now specified as treatment prior to painting include:

1. Projectiles such as rocket war heads, high explosive shells, armor piercing shot
2. Steel boxes or containers for small arms ammunition
3. Fire control instrument components calling for organic finishes
4. Electric and electronics equipment housings.
5. Combat vehicles (Grade I, JAN-C-490 is optional on these according to the current specification MIL-P-3329).

Joint Army-Navy Specification JAN-C-490 is entitled *Cleaning and Preparation of Ferrous Metal Surfaces for Organic Protective Coatings*. The following information has been abstracted for reference by design engineers.

Grade I: Treatment which produces an adherent, crystalline phosphate deposit on a previously clean, ferrous, metal surface.

APPEARANCE OF SURFACE: The appearance of the surface shall be characteristic of the operations performed. Grade I phosphate coatings shall be uniform, fine grained, continuous films free from breaks, scratches, or other damage. A brown or gray streaked appearance which does not impair the adhesion or corrosion resistance shall not be considered cause for rejection.

PHOSPHATE COATING WEIGHT (Grade I only).—For metal parts treated by immersion or spray processes, the minimum weight of phosphate water-insoluble coating shall be 150 mg per sq ft of surface area. Phosphate-coated sheet metal intended for fabrication after painting shall have not less than 40 nor more than 100 mg of phosphate deposit per square foot of surface area.

SALT-SPRAY RESISTANCE (Grade I only)—Grade I shall withstand salt spray test for 250 hours when coated with 0.7 to 1.2 mils of enamel conforming to U. S. Army Specification 3-181, type II, or zinc chromate primer conforming to Army-Navy Aeronautical Specification AN-TT-P-656 as applicable and as specified by the inspector, without the appearance of rust, softening or blistering of the enamel beyond ⅛-inch on either side of the scratch, and without any evidence of film failure such as blistering or corrosion at other points.

USE: The methods described are intended for use in the preparation of iron and steel components for the application of organic coatings (paint, varnish, lacquer, enamel, etc.). They are not intended for mechanical, automotive, or other equipment or material for which there are applicable specifications.

FLASH ZINC-PLATED, PHOSPHATE-COATED SHEETS will be used when required by the drawing or specification for the component or when approval is obtained from the contracting officer. A flash plating

of zinc prior to phosphate coating is not covered by this specification.

PHOSPHATE COATING (Grade I): The properly cleaned articles should be subjected to a balanced phosphate solution, containing nitrate as an accelerating agent (or other phosphate solution which is known to accomplish equivalent results), until a uniform crystalline phosphate coating is obtained. The article should then be rinsed in clean water, followed by a second rinse in a dilute chromic-acid solution and dried.

RINSING: Failure to remove water soluble cleaning compounds by rinsing, while its harmful effect on the paint coating is not immediately apparent, will result in early failure of the paint by blistering, flaking and rapid spread of rust from a scratch. The processes of Grade I, should always be followed immediately by:

1. A clean hot (180 F to boiling) water rinse with a constant overflow maintained by the continuous addition of fresh water.
2. A final hot (160 to 210 F) rinse maintained at a pH of 3 to 5 through the addition of flake chromic acid or proprietary mixtures of chromic and phosphoric acids. This final rinse should be checked by a standard free and total acid titration at least once every 3 hours and the bath discarded when the total acid reading rises to more than three times the free acid reading.

TEMPERATURES: In general, it will be advisable to avoid metal temperatures above 130° F for cellulose lacquers or 160° F for enamels, unless coating-material has been specially formulated for the purpose.

The production sequence involves five or six stages of processing in an automatic power spray washing machine through which the parts are carried on an overhead conveyor, or dipping the part in tanks, or, more recently, by brushing with portable hand equipment. A typical sequence is as follows:

1. Precleaning the metal by spraying it with an emulsion-alkaline cleaner
2. Rinsing with clean water, occasionally two stages
3. Coating the metal in a zinc-phosphate solution
4. Rinsing with clean water
5. Rinsing with chromic or phosphoric-acidulated water.

When products are of the same general design, constructed of cold-rolled steel, and adapted to processing on a monorail conveyor or belt, a multistage power washer is the most practicable equipment to use whether the products are large like automobile or bus bodies, or relatively small like radio cabinets. For products small enough to be processed in tanks singly or on racks, in crates, or in tumbling barrels, dip phosphate coating processes are indicated. For rapid production, automatic equipment can be used to mechanize the line. For coating assembled structures of cold-rolled, as well as hot-rolled steel, such as bridges and gas holders that cannot otherwise be conveniently processed, and for the refinishing of automotive, railway, and other sheet steel equipment a portable brush, spray or flow process of application is recommended.

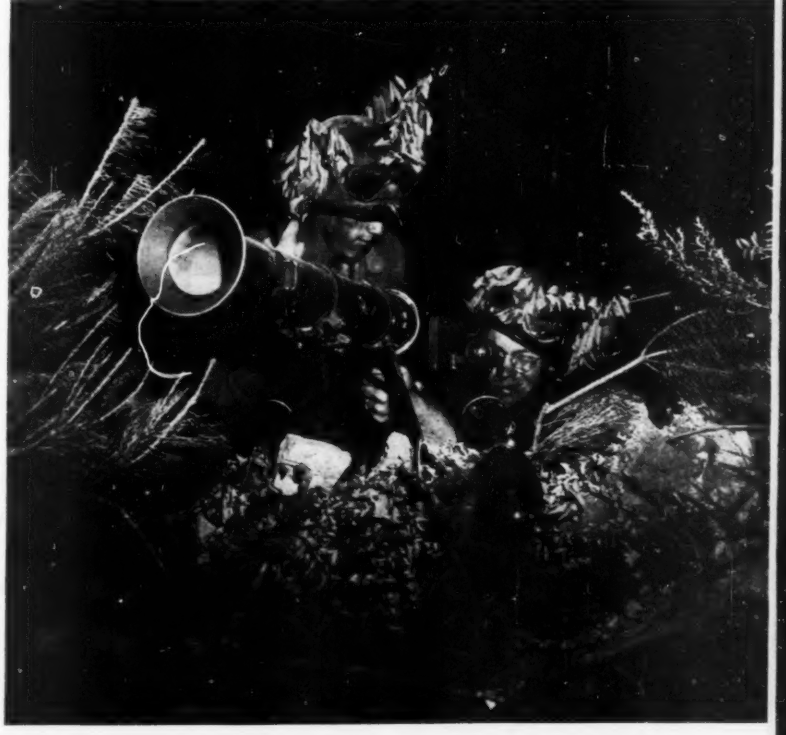
Zinc phosphate coating prior to painting accomplishes these important objectives:

1. It alters both the structure and the nature of the metallic surface subjected to treatment. The action of the coating chemical on metallic iron transmutes the surface into a layer consisting of essentially zinc phosphate. This protective integument is crystalline and nonmetallic; and is integrated with the parent metal.
2. Finish life on phosphated steel is longer than that on untreated steel.
3. The phosphate coating tends to suppress the deleterious activity arising from galvanic corrosion. Rust or corrosion is restricted to areas where accidental scratching, denting or chipping has exposed bare metal. The spread of rust or corrosion beyond the locale of damage is greatly inhibited and retarded by the presence of a phosphate bond under the paint finish.
4. The phosphate coating is uniformly structured. When paint is applied it flows into the regularly patterned crystalline interstices and is there firmly "keyed" or anchored on drying.
5. High abrasion and impact resistance are imparted to the finish on zinc-phosphate coated steel.
6. High luster of the final finish is typical of refrigerator and automotive finishes applied over zinc phosphate coated metal.
7. The hard, dense and smooth coatings produced by the chemical action of zinc phosphate on steel has resulted, in a number of instances, in reduced paint consumption.

Improving Paint Adhesion on Aluminum

The expanding use of aluminum in industry has given rise to several problems in painting and surface protection, notably that of assuring permanence of adhesion under the widest possible variety of service conditions. It is not always appreciated that the adhesion of paint to steel or tin plate, for example, involves a series of physical conditions and a mech-

Firing a 3.5 rocket launcher. Aluminum parts are treated with a chemical film, primed with zinc chromate and painted with cellulose nitrate lacquer



anism different from those encountered with aluminum. Thus, while the use of aluminum may eliminate a rusting problem, difficulties and disappointments may arise if the essential surface differences between ferrous metals and light alloys are ignored.

Aluminum not only corrodes when exposed unpainted to the atmosphere (particularly in moist, salt-laden air or industrial fumes) but also sheds paint unless the chemical nature of the surface is actually changed prior to finishing. Simple treatments involving cleaning, etching, or both, which heretofore have been used extensively, do not change the chemical composition of the surface and are inadequate. Far from retarding the corrosion of unpainted aluminum, such processes may in fact accelerate it.

Solvent cleaning, for example, removes oil completely without etching the metal. Phosphoric-acid solvent or chromic-acid cleaners, both commonly used, remove oil completely and, in addition, minutely etch the aluminum surface. However, none of these treatments appreciably improves the bond or the protection of the paint.

Alkalis have long been favored by many metal finishers on the theory that oil is removed and the surface etched as deeply as desired to provide a good "tooth" for paint. Alkali cleaning, however, has proved unsatisfactory.

Integral Coatings Are More Effective

In general, coatings integral with the aluminum itself have proved to be far more effective than cleaning and etching for bonding paint and enhancing its protection. These coatings fall into three main categories depending on their composition: conventional crystalline-phosphate coatings; oxide coatings; and the newly developed amorphous-phosphate coating.

While the conventional crystalline-phosphate coatings afford a good "tooth," they provide little protection for the unpainted metal. Better protection is obtained with oxide-forming processes. However, these are expensive and time-consuming. This is especially true of electrolytic oxidizing (anodizing) which until recently was generally accepted as the best protection for aluminum.

The ideal pretreatment for aluminum will do three things:

1. Free the surface from all traces of contamination and corrosion (cleaning)
2. Protect the metal from corrosion by forming an impervious inert layer on the surface (coating)
3. Provide a sure and durable key for paint (coating).

In general, coatings integral with the aluminum itself have proved to be far more effective than merely cleaning and etching for bonding paint and enhancing its protection.

Chemical films such as the newly developed Alodine protective coatings covered by Military Specifications MIL-C-5541 and MIL-S-5002 provide an even better bond and more durable protection for paint than either conventional phosphate coatings or the laborious oxide forming (anodizing) processes. These protective coatings are used to obtain a finish on military aluminum products such as: Naval aircraft, electronic equipment, cartridge, powder and rocket tanks of aluminum, ground signal equipment, and other aluminum parts and products.

The process is a chemical one and consists of the following easily controlled operations or steps:

1. Cleaning the work
2. Rinsing the cleaned aluminum surfaces
3. Coating with the phosphate
4. Rinsing with clean water
5. Rinsing with warm acidulated water.

These steps are followed by drying and painting operations.

Parts can be treated by immersion in tanks, spraying in an industrial washing machine, flow coating, or brushing. In general, small-size products or parts are processed rapidly and conveniently in immersion equipment which can be mechanized if production volume justifies it. For large production of formed parts, or for processing coiled stock, strip or cut-to-size sheet, a power spray washer is convenient.

Abstracts from Military Specification MIL-C-5541, *Chemical Films For Aluminum And Aluminum Alloys* follow:

WORKMANSHIP: The chemical films covered by this specification shall be produced by suitable treatments controlled and operated to give a uniform product conforming to the requirements specified herein.

1. The film shall be continuous and free from breaks, scratches, and other damage within the limits of best aircraft practice.
2. All details of workmanship shall be in accordance with the best aircraft practice covering this class of product.

USE: Chemical films covered by this specification are intended for use as a paint base and as a corrosion preventive film for aluminum and aluminum alloys.

1. *Chemical films* produced under this specification may be used in lieu of anodic films, Specification AN-QQ-A-696, when specifically authorized by the Procuring Service.
2. *Test Specimens:* Particular caution must be exercised to insure that test specimens submitted for corrosion testing are not susceptible to intergranular corrosion since such susceptibility will cause premature failure.

The advantages of chemical surface treatment such as Alodine to product designers are as follows:

1. Protection imparted to unpainted aluminum compares favorably with chromic anodizing and, as a preparation for paint, the treatment has been found to be superior. There have been numerous salt spray tests of 2,000 hours and more without failure of coated and painted aluminum
2. Adhesion of the coating to the metal is perfect. Aluminum can be bent flat and destruction of the metal will invariably precede that of the coating
3. Coatings will withstand heat that will melt the aluminum
4. No appreciable dimensional changes occur when aluminum is coated
5. Coated aluminum shows good resistance against bimetallic or galvanic corrosion.

Aluminum and its alloys are now being specified for more and more military items. The slogan "Make

(*Continued on Page* 196)

By W. A. Thomas
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Chemical Plants Division
Blaw-Knox Construction Co.

Determining Natural Vibration Period

. . . of compound-section cantilever beams

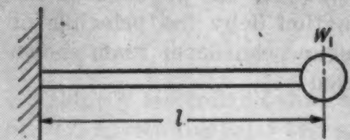


Fig. 1 — Theoretically weightless cantilever beam having load mass concentrated at the free end. Period of vibration is determined by using Equation 1

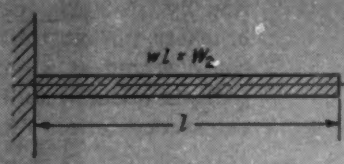


Fig. 2 — Cantilever beam having uniform cross-section, and real uniform weight throughout. Period of vibration is determined by using Equation 2

IN DESIGNING machines of various types, it is often necessary to determine the vibration period of a part in the form of a cantilever beam. Unfortunately, the beam is not usually an ideal cantilever. More often it is one which has abruptly changing sections and, therefore, abruptly changing moments of inertia. A typical and common example is that of a tall stack or tower, which will usually have not more than three different cross sections. A direct general formula is developed in this article to aid in calculating the vibration period of the three-step beam. It may be easily extended to cover cantilever beams having more than three different cross sections. In *Fig.* 1 is illustrated a theoretically weightless beam which is considered to have a real moment of inertia, I, in.^{4} At the end of this beam is a weight, W_1, lb. The period of vibration of the weight is, in seconds, approximately[1],

$$T_1 = \sqrt{\frac{0.034 W_1 l^3}{EI}} \qquad (1)$$

In *Fig.* 2, a cantilever beam has a real uniform weight W_2, lb, or w, lb per inch, but is not otherwise loaded. The period of vibration[2] is, approximately,

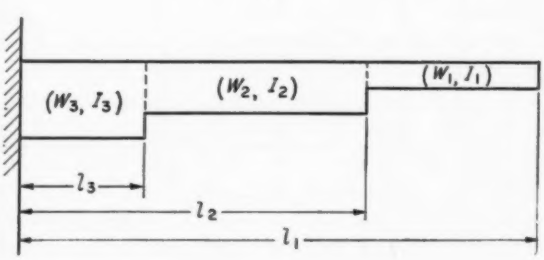
$$T_2 = \sqrt{\frac{0.0082 w l^4}{EI}} = \sqrt{\frac{0.0082 W_2 l^3}{EI}} \qquad (2)$$

Assuming that E, modulus of elasticity, psi, I, and l, in., are the same for both beams, by equating Equations 1 and 2, there results $W_1 = (\frac{1}{4})\ W_2$, approximately, showing that the beam illustrated in *Fig.* 2 vibrates with the same period as the beam shown in *Fig.* 1 when $W_1 = \frac{1}{4}W_2$. Accordingly, by the principle of superposition for a combination of the loadings of both beams, the period is given by

$$T_3 = \sqrt{\frac{0.034(W_1 + \frac{1}{4}W_2)l^3}{EI}} \qquad (3)$$

In *Fig.* 3 is shown a compound cantilever beam having three different cross sections. W_1, W_2, W_3, and

Fig. 3—Cantilever beam of compound section. Equation 4 is used to determine the natural vibration period of this type beam



[1] References are tabulated at end of article.

l_1, I_2, I_3, representing the total weights and the moments of inertia, respectively, of the portions of the beam. Extension of Equation 3 by the principle of superposition to the compound cantilever beam shown in *Fig.* 3 results in the period

$$T = \left\{ \frac{(W_1 + W_2 + \frac{1}{4}W_3)\, l_3^3}{30EI_3} + \frac{(W_1 + \frac{1}{4}W_2)\, l_2^3}{30EI_2} + \frac{\frac{1}{4}W_1\, l_1^3}{30EI_1} \right\}^{\frac{1}{2}} \quad (4)$$

In deriving Equation 4, the weights W_1 and W_2 are considered as being concentrated at a point l_3 distance from the support, and $\frac{1}{4}$ W_3 is added for the distributed weight of the beam up to l_3. Multiplication by $l_3^3/30EI_3$ results in the first term within the brackets. Next the load W_1 is considered as being concentrated at a point l_2 distance from the support, $\frac{1}{4}$ W_2 is added. Multiplication by $l_2^3/30EI_2$ results in the second term. For the third term there is only the distributed weight $\frac{1}{4}$ W_1 multiplied by $l_1^3/30EI_1$.

Equation 4 satisfies the limiting conditions that when any two of the three lengths of the cantilever are zero, it reduces to Equation 2. Also when $W_1 = W_2 = W_3$ and $l_1 = l_2 = l_3$, the period t is the same as that by Equation 2 in which l is to be equal to $l_1 + l_2 + l_3$ of Equation 4.

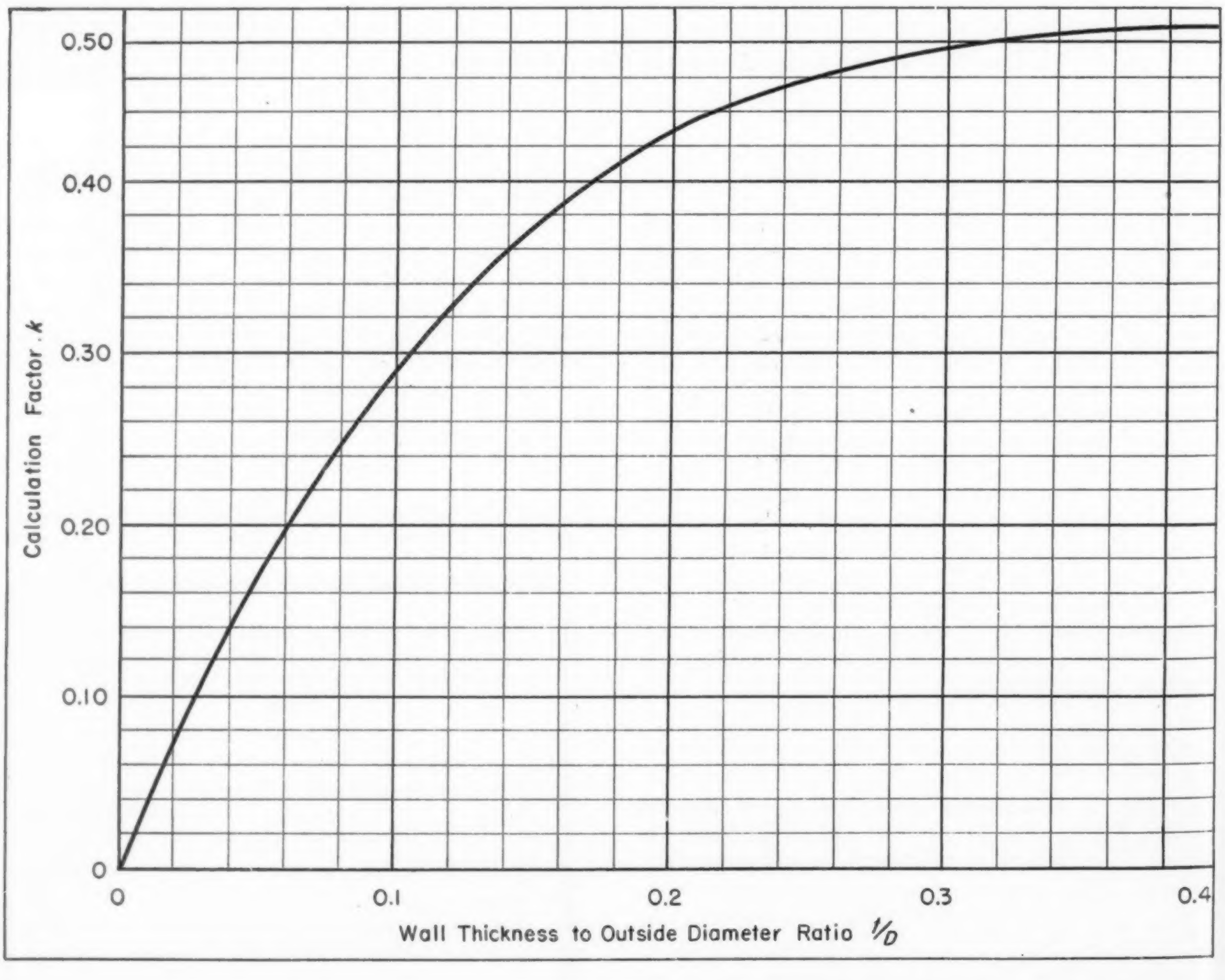
Since stepped hollow circular sections represent a commonly occurring case in the practical application of Equation 4, it will be of additional utility to have a rapid means for determining the moment of inertia of such a cross section.

For a hollow circular section the moment of inertia about the transverse central axis is, precisely

$$I = \frac{\pi(D^4 - d^4)}{64} \quad (5)$$

where D is the outside diameter, inches, and d is the inside diameter, inches. Replacing d with t, the

Fig. 4—Graph of calculation factor versus ratio of wall thickness to outside diameter of tubular beams. Factor k enables rapid solution for moment of inertia, $I = kD^4$



thickness in inches, results in:

$$I = \frac{\pi[D^4 - (D - 2t)^4]}{64} \quad \text{(6)}$$

Expanding Equation 6 and rearranging terms results in

$$I = 0.3927\, D^4 \left[\frac{t}{D} - 3\left(\frac{t}{D}\right)^2 + 4\left(\frac{t}{D}\right)^3 - 2\left(\frac{t}{D}\right)^4 \right] \quad \text{(7)}$$

Rearranging Equation 7 for convenience, $I = k\,D^4$ in which

$$k = 0.3927 \left[\frac{t}{D} - 3\left(\frac{t}{D}\right)^2 + 4\left(\frac{t}{D}\right)^3 - 2\left(\frac{t}{D}\right)^4 \right] \quad \text{(8)}$$

Factor k is available directly from the curve shown in *Fig.* 4. As the ratio t/D approaches 0.5, the k factor becomes more nearly constant. Between t/D ratio of 0.4 and 0.5, factor k may be taken as 0.495 with a maximum error of one per cent.

REFERENCES

1. John Goodman—*Mechanics Applied to Engineering*, Vol. 1, Longmans Green & Co., London, England, Page 264.
2. *Kent's Mechanical Engineers' Handbook, Design and Production Volume*, Twelfth Edition, John Wiley & Sons Inc., New York, Page 9-03.

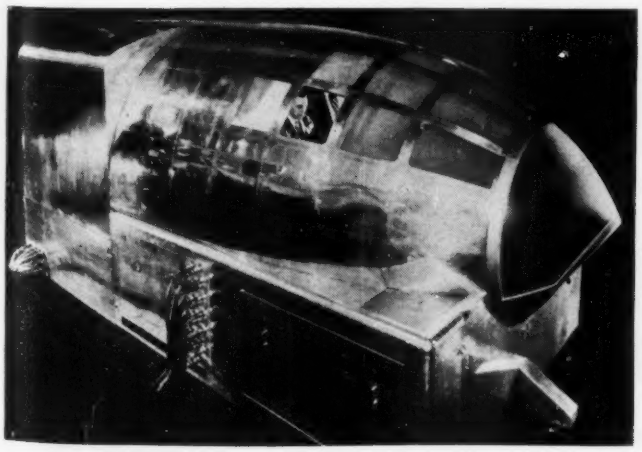
Complex Servos Operate Flight Trainer

AIR FORCE pilots of the future will soon be trained on new flight simulators which duplicate the cockpits of huge B-47, B-36 and B-50-D bombers, C-124 and C-97 cargo planes, and the F-86-D all-weather interceptor.

Up to now only a one-station simulator was available to the Air Force, the Link Jet Trainer. The new device, built as either a one, two or three-station trainer, will contain controls for the pilot, co-pilot and flight engineer where applicable, and will give training on a specific airplane.

Besides flight attitude, the simulator can duplicate engine fires, and fuel-system, electrical-system, instrument, and landing-gear failures. A pilot entering a simulator finds an exact duplicate of the cockpit of the plane for which it was designed, with one exception—the windshield is painted shut. As he goes through the various pre-flight checks, the instruments, operated by electronic computers, reflect the condition of the simulated airplane, just as they would on the aircraft itself.

If the pilot makes a mistake, the computers cause the instruments to conform to the actual condition that would be present in an airplane. Unless the mistake is corrected the condition will go to its conclusion—either a burned out tail-pipe, fire in an engine, or whatever might be the result of that mistake.



Working heart of the device consists of analogue computers. An array of servo mechanisms and amplifiers instantly, automatically and continuously solve differential equations relative to flight and engine power. A computer for an engine, for instance, reveals the outside air temperature, revolutions per minute, manifold pressure, cowl-flap setting, oil-cooler setting and intercooler setting.

All these servos are related to each other, and many receive their controlling impulses from four, five or six other units. As an example, when the flight engineer opens or closes the cowl flaps for any engine, the cowl-flap position is indicated on his instrument panel and, in addition, a signal is transmitted to the engine cabinet. The cowl-flap servo then sends out impulses which vary the airplane's drag, yaw, attitude and airspeed, and the temperature of that particular engine.

Conveyor Removes Steel Scrap

A 236-FOOT run of hinged-steel conveyor belting has eliminated manual scrap removal from a line of nine 250-ton punch presses at the Midland Steel Products Co. The highly abrasive sheet-steel scrap passes down a chute from each machine onto May-Fran belting which is driven by a variable speed drive at 20 to 40 fpm.

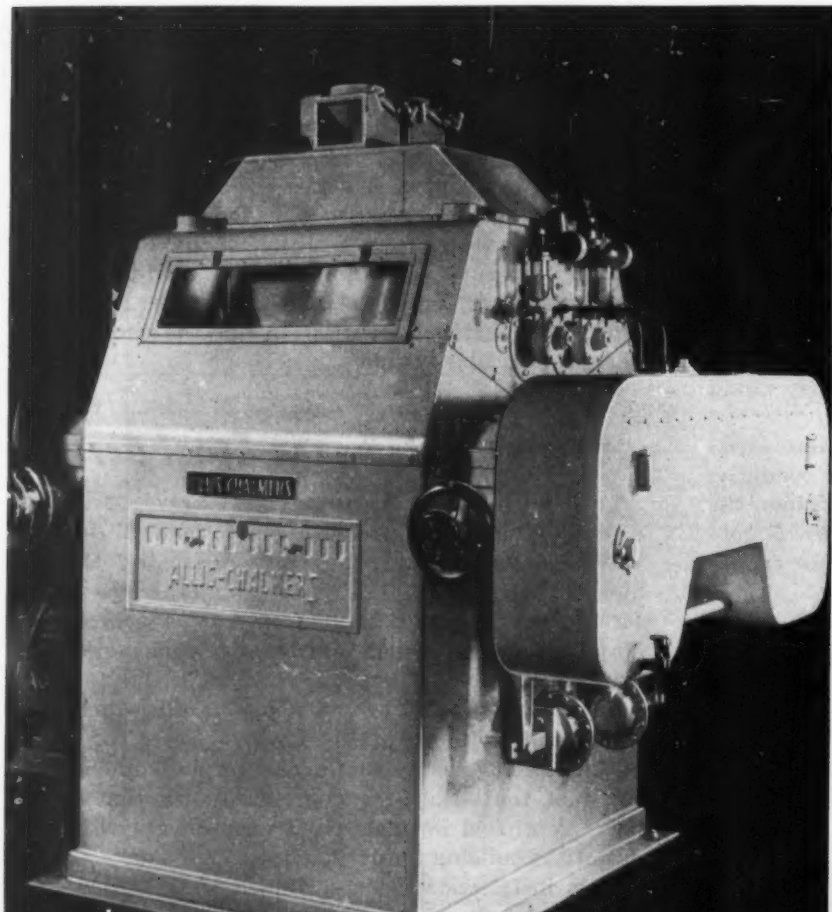
The belting feeds the scrap into an alligator shear, which cuts it into 6 to 10-inch long pieces for better packing. To assure continuous scrap handling, the speed of the shear is synchronized with that of the belt. A 21-foot inclined unit transports the cut scrap out of the building for car loading.

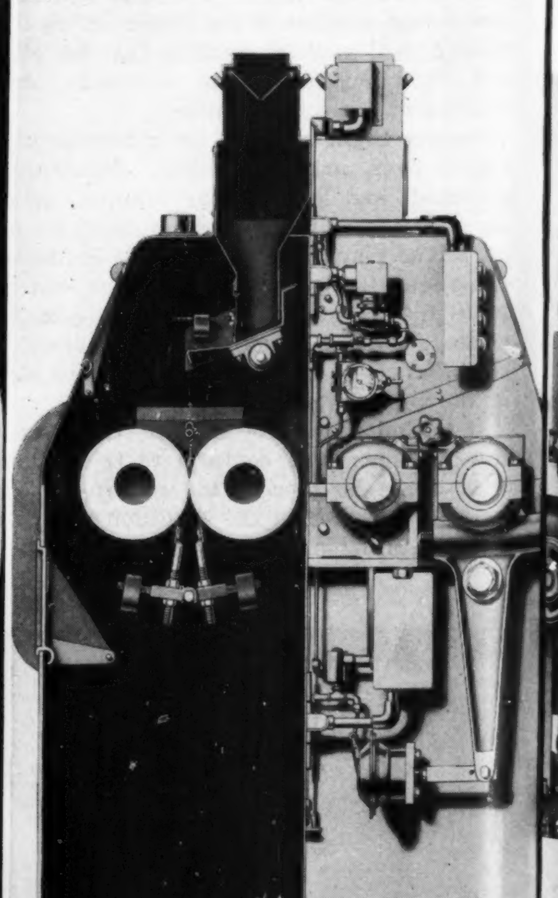
This installation utilizes 36-inch wide belting with solid steel links of 6-inch pitch, assembled with high-carbon steel rods. The side chains are an integral part of the belt. Outside links incorporate interlocking wings which remain engaged at all times, even when the belt is traveling from one level to another or over power and idler sprockets.

CONTEMPORARY

New Roller Mill Has Automatic Feed a

BETTER feed distribution and uniform pressure at both ends of the milling rolls result in more efficient operation of the grain mill shown below. Being driven by air motor, a high-speed vibrating feeder with variable-speed adjustment provides accurate feed regulation consistent with roll gap setting. Predetermined gap adjustment is accomplished by manual operation of calibrated handwheels, a minimum gap being allowed by spacers arranged to prevent metal to metal contact of the roll surfaces. Eccentric adjustment of the operating lever fulcrums enables parallel alignment of the rolls to be attained initially. Diaphragm type cylinders are employed to shift the movable rolls into operative position, and maintain equalized roll pressure thereafter during the milling process. Feeder and roll controls are interlocked so that the rolls separate automatically when the feed is stopped. This action is initiated by Mercoid switch operation caused by self-closing action of the feeder gate when the feeder mechanism is stopped. When the mill is again set in operation, the rolls are permitted to attain full speed

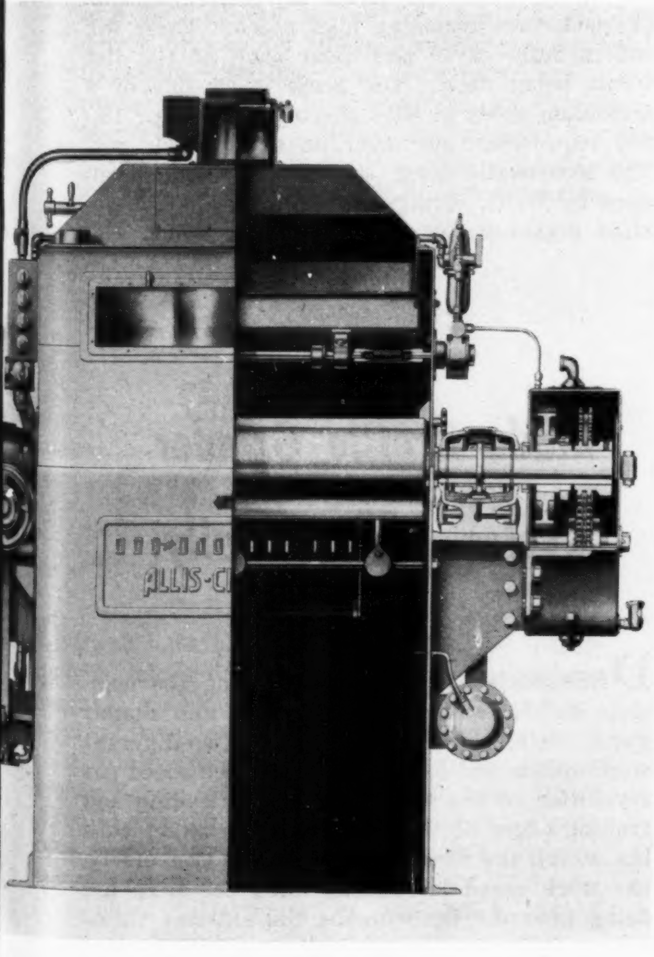


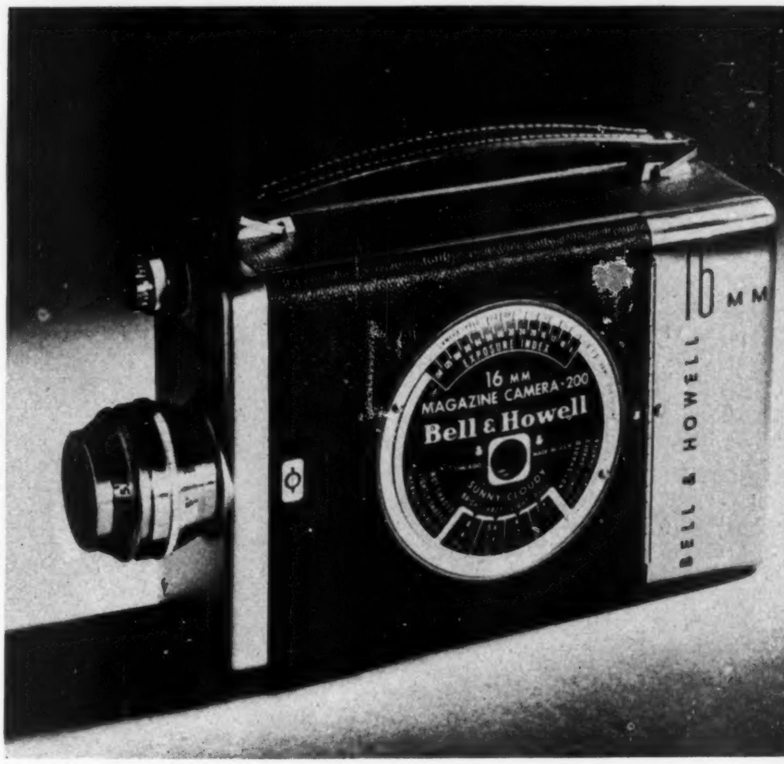


Y DESIGN

al Control

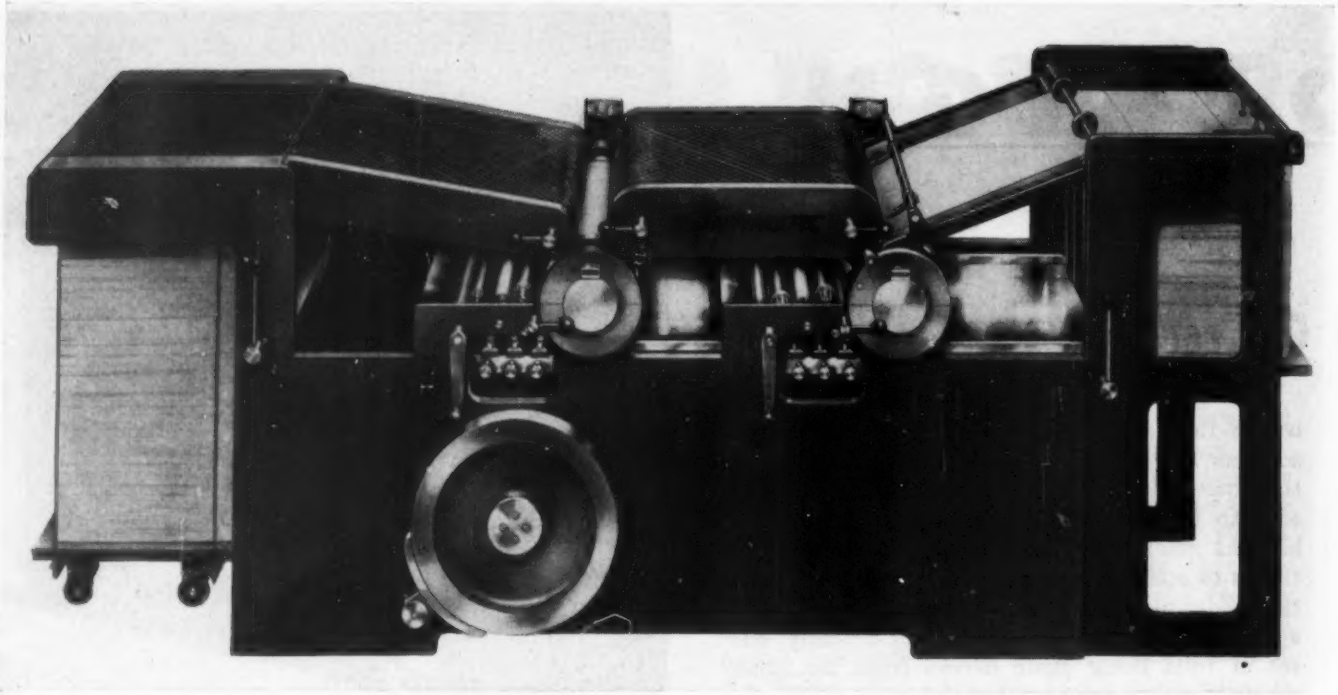
before the feeder motor starts to operate. As soon as flow into the feeder causes opening of the flow gates, the rolls are automatically shifted into milling position where they remain as long as stock flows into the feeder and it continues to operate. If the roll vibration is stopped the feeder also stops, and the rolls automatically separate. Each mill is a dual unit actually, one set of rolls being chain driven from the other which is motor driven. Roll pairs are geared together by hardened helical gears, and reduced speed of the secondary rolls is possible with suitable sprocket combinations. The AirSet roller mill, built by Allis-Chalmers, is said to be the closest approach yet to automatic milling.





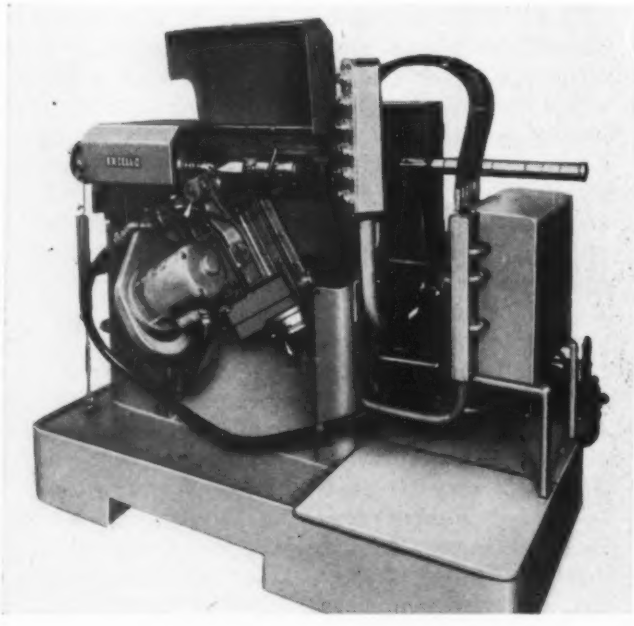
Award Winning Movie Camera

WINNER of the 1952 Society of Motion Picture Art Directors Award is the 16 mm magazine camera shown above. New design features include a simplified built-in exposure guide which is highly legible and convenient; a leather covered spring steel handle that lays flat when not in use; and a subtly curved case of diecast aluminum body having gray sculptural finish and satin chromium trim. A full-length continuous type hinge provides sturdy mounting of the end cover to the case. Optical features include a two-lens turret with automatically positioned positive type matched viewfinder objectives. Loading time is 3 seconds, and exposure speeds are 16, 24, 32, 48 and 64 frames per second. The camera is equipped with a single frame exposure and starting button lock. Black and white or color film may be used, and the magazine can be loaded without removing the tripod. The Bell and Howell "200" was designed by Reinecke and Associates, Chicago, in collaboration with Bell and Howell engineers.



Two-Color Press Has High Speed

THIS new flat-bed printing press, above, embodies features which give it approximately twice the speed of existing presses, size for size. The operating technique for the two-color machine is the same as for a single color flat-bed machine. The press is equipped with an automatic stock feeder and an ingenious transfer unit which delivers the sheet in perfect register from the first cylinder to the second. Rotation of the cylinders is fully synchronized with the bed travel, slowing down at the end of the stroke when the sheet is received or discharged, then resuming high speed. Three ink rollers fully cover and clear each of the flat forms being used. The press is capable of a maximum speed of 8000 sheets per hour, or 16,000 impressions per hour on a two color run. The Montimatic press was designed and developed by W. G. Montgomery, Marietta, Ohio, retired president, Miller Printing Machinery Co.

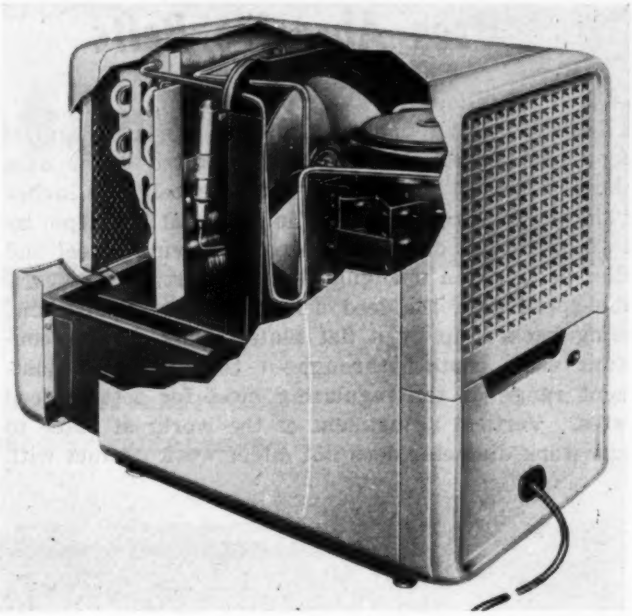


Blade Profile Grinder Cycles Automatically

DESIGNED and built for grinding the complete airfoil form of turbine blades and similar parts, the machine shown, left, has two different work speeds and four changes of work speed per revolution of the work. When the leading and trailing edges of the blades are in contact with the wheel, the work speed is high. Conversely, the work speed is reduced when the sides are being ground. By reducing the extreme varia-

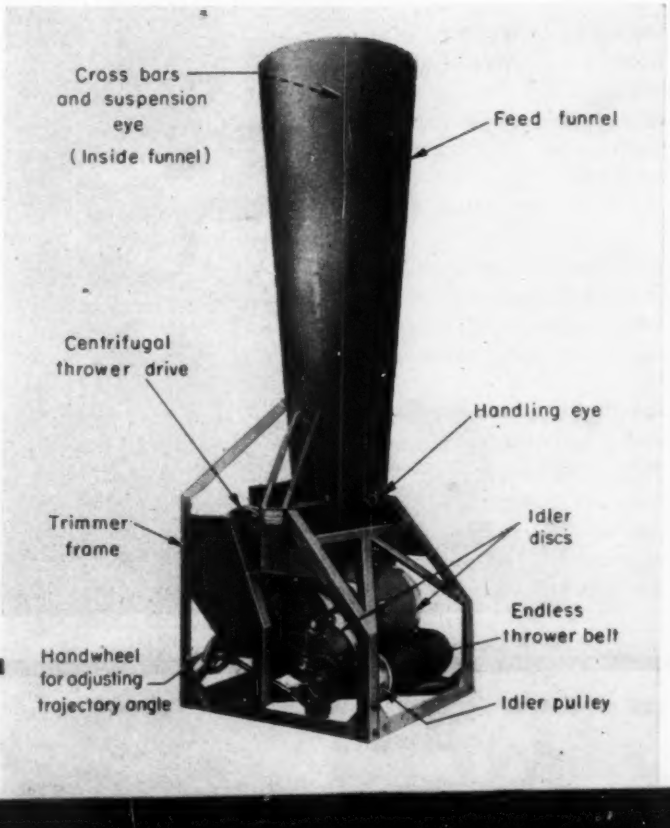
tions in work surface speed that would exist if work rpm were constant, a better surface finish quality is thereby obtained. The work oscillation relative to the wheel is controlled by an accurately profiled master cam. After the blade has been loaded in the machine and the cycle start button depressed, the machine proceeds in automatic cycle, first grinding the fillet and shoulder at the root of the blade, then grinding the contour at a controlled traverse speed rate toward the tip of the blade. At the end of the traverse grind operation, the machine resets rapidly to await reloading. Built by the Ex-Cell-O Corp., the machine is self-contained, being complete with hydraulic and coolant reservoirs, pumps, and electrical control equipment.

Compact Dehumidifier Easily Portable



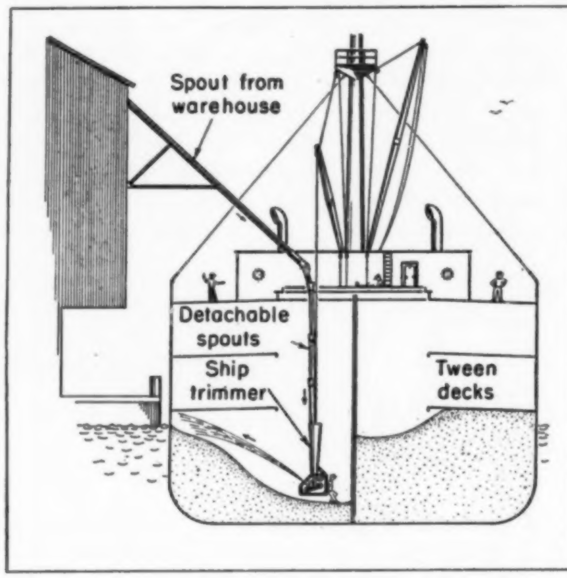
PORTABLE unit capable of dehumidifying up to 8000 cubic feet of atmosphere without materially increasing the room temperature has been designed by engineers of General Motors Co., Frigidaire division. Included in the 2.5 cubic feet of space occupied by the unit shown, right, is a cleanable air intake filter, and a drawer type moisture receptacle with a capacity of over 22 pints. Convenient hand-holds are provided at the end of the shell, which rests on four leveling glides. Air circulation at the rate of 165 cfm is produced by an 8½-inch 4-blade fan driven by a motor which is rubber-pad mounted to isolate vibration. The ⅛-hp refrigerant compressor is similar to the well known Frigidaire Meter-Miser unit. Styled by Raymond Loewy, the unit is simple to operate, requiring only service cord plug-in and toggle switch actuation.

Ship Trimmer Cuts Loading Time



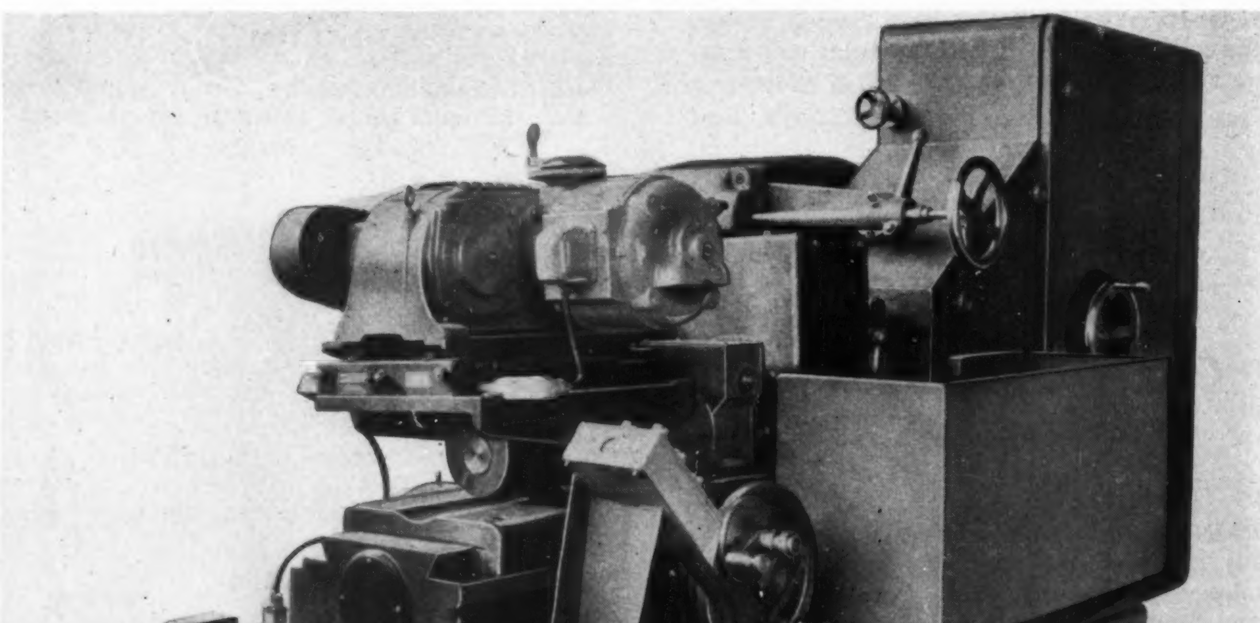
GRAIN and other bulk materials are loaded rapidly and uniformly into ship holds by the portable machine shown, right. Operating principle of the trimmer is simple. The material is chuted into the unit onto an endless belt that is motor driven at a speed of 2700 fpm. Idler disks at the outer edge of the belt form a concave pocket from which the material is projected approximately 50 ft from the trimmer. Handwheel adjustment of the idler pulley

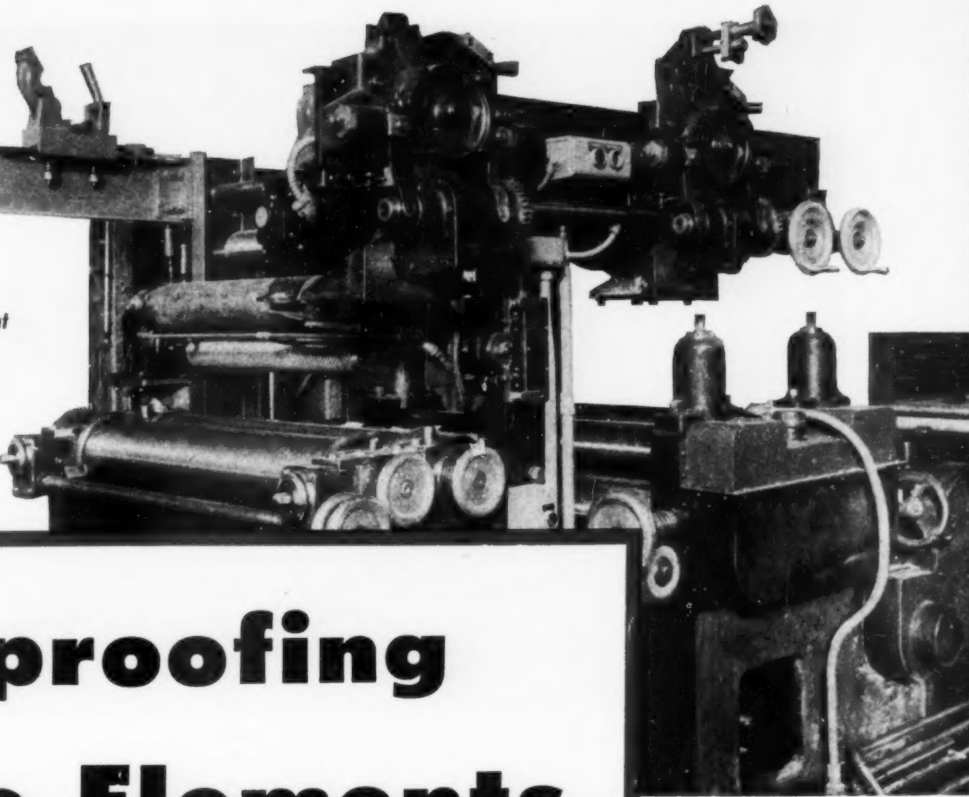
changes the trajectory angle of the stream, enabling close control of material distribution throughout the hold, as illustrated, right. Manufactured by Stephens-Adamson Mfg. Co., a fixed type trimmer is also made for permanent mounting to boom type belt conveyors. Capacities range up to 1500 tons per hour, and in most cases, manual attention involved is about 20 per cent of that required by other loading methods.



Centerless Grinder Uses Abrasive Belts

INSTEAD of the conventional grinding and regulating wheels, the new machine illustrated below uses abrasive belts. The endless grinding belt is 9 inches wide by 168 inches long. It is driven at 8500 fpm by a 25-hp motor, and runs over the driving wheel and idler pulleys in a triangular course. Driven by a variable-speed unit, the feed belt runs in vertical plane, being backed up by a flat plate. This enables constant work contact throughout the angular adjustment range of the regulating head for setting feed rates. Vertical adjustment of the workrest blade to suit work diameter does not affect work contact with the feed belt as with the conventional regulating wheel. These features enable quick setup changeovers in this centerless grinder since the slow process of regulating wheel truing is not required. Built by the Production Machine Co., the machine may be used for wet or dry grinding and finishing cylindrical work ranging from ½ to 6 in. diameter, by both infeed and throughfeed methods.





By Paul B. Berlien
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Wearproofing Machine Elements

. . . with cemented carbides reduces machine operating and maintenance cost, increases customer acceptance

WHAT does a machine cost the user? There is of course the initial cost. But to this must be added the cost of replacing worn parts during the machine's useful life, *plus* the cost of productive time lost (sometimes for an entire department) when the machine is down for repairs, *plus* the cost of material spoiled when the machine is not operating properly. As a result, many machines actually cost several times the amount shown on the original price tag. If the designer could remove more of the upkeep costs *before* a machine goes to work, tremendous advantage in time, labor, materials, and customer acceptance would result.

One of the primary factors which builds upkeep costs, of course, is not out-and-out breakage, but ordinary wear. To combat this factor, the designer should actively work to eliminate costly wear in critical machine elements wherever it may possibly or probably occur.

A simple plan that has proven highly effective is to design or redesign so as to "wearproof" those areas of machine components which will be subject to high wear loads. Wearproofing materials employed for this purpose include the well-known cemented carbides. Machine designers are apt to think of cemented carbides only as tool materials. Actually, their extreme wear-resistant properties, plus their ready availability at low cost in many standard shapes and sizes, make them ideal materials for wearproofing machine elements.

Following are some examples of uses of carbide in wearproofing basic machine parts, together with data on how the carbide was applied. It is hoped that these examples may suggest similar and other applications to machine designers.

Guides: The simple application shown in *Fig.* 1 resulted in a reduction of machine maintenance costs from $1307 to $109 per year. A cemented carbide ring was substituted for cyanided steel or ceramic eyes formerly used to guide the wire in a de-reeling machine. The carbide guides are silver brazed directly to the arm instead of bending the arm around the guides as with ceramics.

In another case, a guide ring of cemented carbide

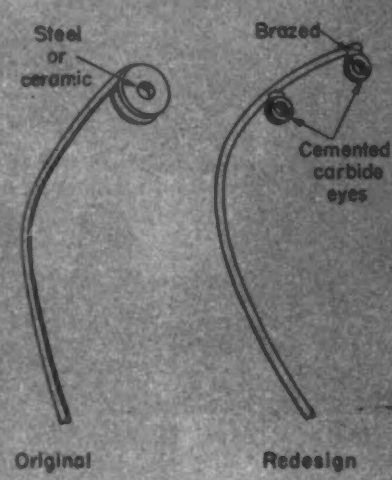


Fig. 1—Left—In a wire de-reeling machine, Carboloy cemented carbide guides were substituted for the hard steel or ceramic guides formerly used. This simple wearproofing idea resulted in a reduction of 91 per cent in the cost of repair and replacement formerly required for the guide

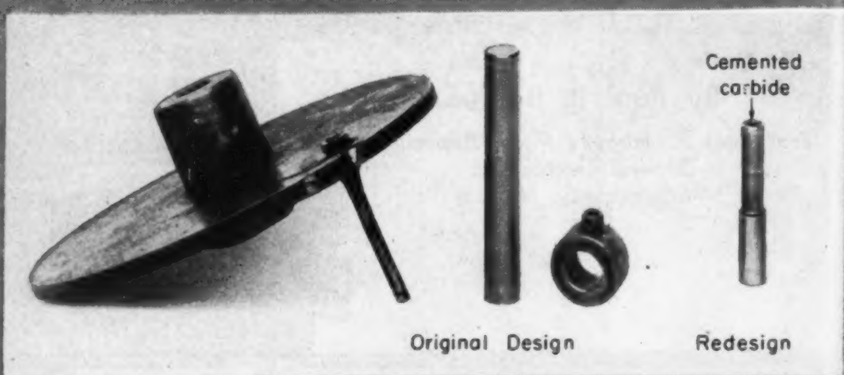


Fig. 2—Above—In order to make repair and replacement easy on a part that quickly wore out, a nine-piece assembly was employed. By redesigning with a carbide insert the part could be mounted direct in a major machine element and the nine-piece assembly eliminated

not only reduced maintenance expense but also simlified machine design. In this machine, the original guide wore so rapidly that a nine-piece assembly was required so that the part could be removed and replaced quickly, *Fig*. 2. When made of carbide the guide could be mounted directly in a major machine element, the nine-piece assembly being eliminated. The Carboloy insert cost only 13 cents—the savings in repair costs on machines of this type amount to more than $2000 per year.

Cams and Followers: A case where carbide blanks were applied to a cam follower is shown in *Fig*. 3. Originally, the cam followers were designed to be made of bronze and their service life in the machine was only three months. Although this is a small, apparently minor part, the replacement cam shoes on machines of this type were costing some $4000 per year; nonproductive time for repair and setup cost over $2700 per year!

To eliminate these costs the cam followers were redesigned as shown in *Fig*. 3. In three years of constant use the carbide faced shoes have worn only 0.001-inch. They should operate without further attention for the life of the machine.

Another instance concerns a cam with a roller follower. Here the wear occurred on the cam drum itself—an expensive part. The drum cams, *Fig*. 4, are used to position an arm between two cam flats four times during each revolution. As the drum stops at each quarter turn there is a strong side pressure of the roller. When the drum turns again the roller drops off the corner of the positioning flat and rides into the next position. The action had a swaging effect; in time this caused a slap, which in turn increased wear and sometimes would even fracture the cast iron drums.

This cam drum was redesigned to employ four pairs of carbide inserts at the points of wear. Cut from standard Carboloy grade 55A strips, these inserts were brazed into seats milled in the drum at the steps. Each insert is ground on the bearing side only. Total extra cost of this design over the original—labor and materials—was $16, a small fraction of the cost of new drums. Now the drums have a life expectancy equal to that of the machine. Operation is smoother, quieter.

Index Plates: A rather neat solution to a problem of wear on index plates is shown in *Fig*. 5. These index plates required replacement as often as every two or three weeks in service. To redesign these

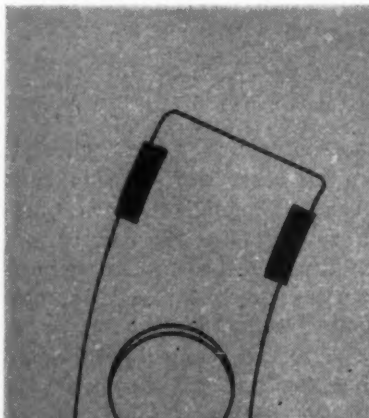


Fig. 3 — Wear on this cam follower was virtually eliminated by the use of carbide inserts. The follower was redesigned to have four inserts brazed in and ground to size on the contacting or wearing faces

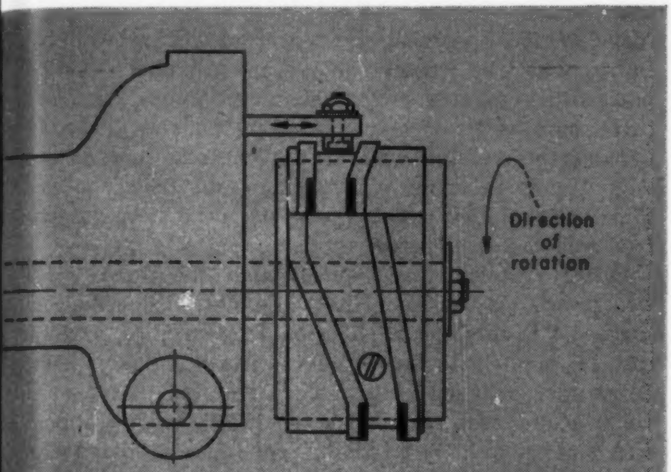


Fig. 4—Wearproofing at the dwell points of this cam was necessary to eliminate faulty machine operation, downtime, repair costs, and replacement of expensive cams. The solution was to employ four pairs of Carboloy Grade 55A inserts at the wear points

units for wear resistance, a cemented carbide face was provided at each of the four stop faces. Ability of the carbide to take repeated impacts has made the plates practically wearproof. In a year of service they have required no attention. There has also been a marked improvement in indexing action and the former progressive loss of accuracy due to wear has been eliminated.

Clutch Collars: Cemented carbide has excellent impact properties when properly used—that is, with body or thickness proportionate to stress. The part illustrated in *Fig.* 6 is a clutch collar reinforced with carbide blanks. The collar is part of a braking assembly within the drum of a machine. While it rotates, a spring-loaded brake slaps sharply against it 240 times a minute. The wear and replacement problem, from the stroking and rubbing action between the collar and the brake, was severe. As redesigned, the inserts have worn only 0.001-inch after three years of service.

The only major design change called for in this part was the milling of grooves to take the blanks (standard Carboloy No. 1240) and chamfering the edges of the carbide blank to eliminate any possible damage from edge chipping.

Toothed Clutches: One way to utilize successfully small pieces of carbide where high reciprocating stresses are present, is to use an extra thick braze to serve as a "cushion" with sufficient elasticity to prevent the carbide blanks from being torn out of the larger body. The application illustrated in *Fig.* 7 gave satisfactory results in this regard. The tangs were formerly made integral with the clutch plate. As soon as wear took place the increased clearance made the tangs subject to continuously repeated high shock loads which in turn aggravated the wear, not only on the tangs but on other machine elements as well.

In redesigning this unit, the tangs were eliminated and slots provided to take cemented carbide inserts. Present service experience indicates that wear on the carbide tangs is almost insignificant. The improvement has been applied to some 250 similar machines.

Welding Heads: One of the most severe abrasive wear problems encountered was in automatic arc welding heads, *Fig.* 8. The nozzles through which the welding wire was fed were originally made of brass

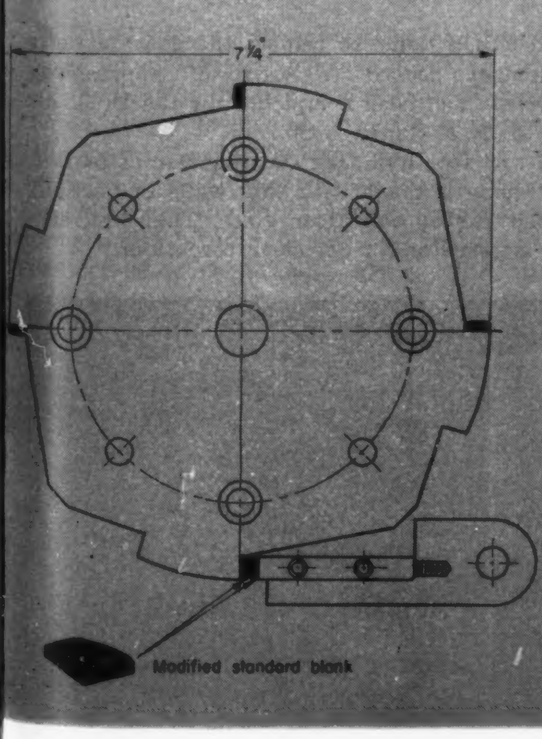


Fig. 5—Left—Redesign solution to the problem of wear and subsequent inaccuracy of operation in this index plate was to provide cemented carbide pads at the indexing surfaces. While former plates often wore out in two or three weeks, the new design has required no attention in the first year of operation

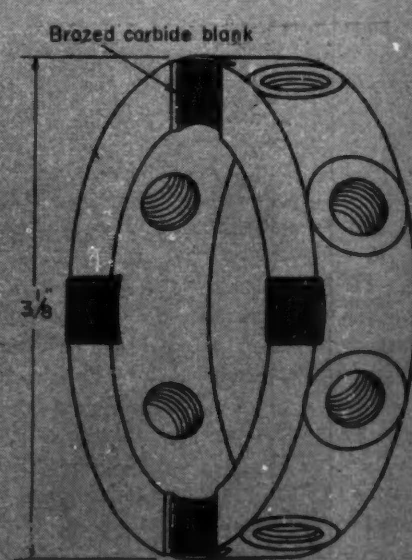


Fig. 6—Above—Though subject to 240 sharp impacts per minute and severe wear, this clutch collar has shown only 0.001-inch wear in three years of use since wearproofing with cemented carbide inserts

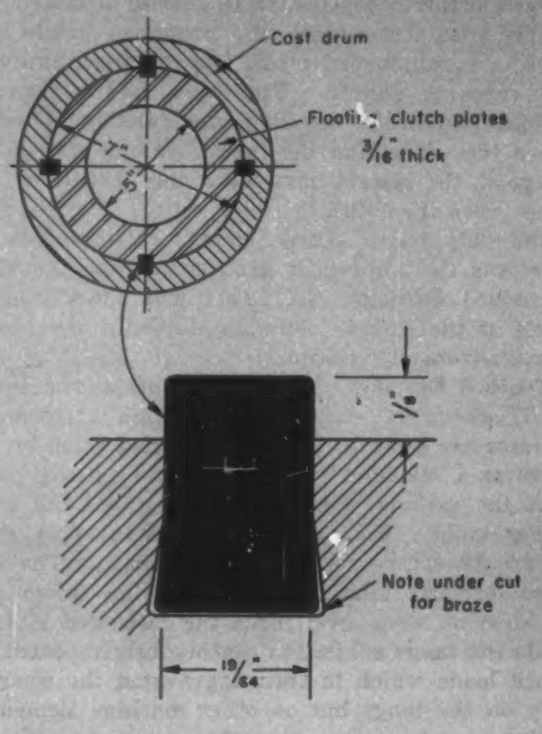
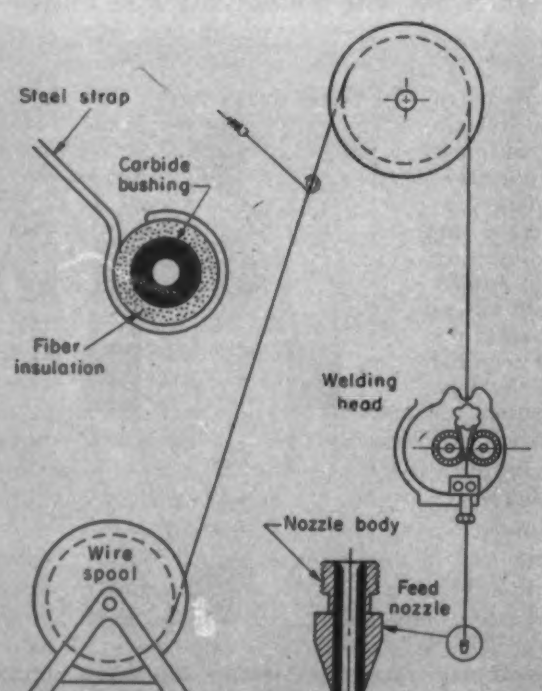


Fig. 7—Above—Clutch plates shown were formerly scrapped when the tangs wore out in 9 to 10 months of service. With the tangs replaced by Carboloy inserts wear is so low that it is negligible

Fig. 8—Below—With anticipated life only two to three weeks, conventional brass nozzles were redesigned to utilize a Carboloy bushing, extending life expectancy to as much as one year



and burned or wore out in about three weeks. To redesign these nozzles, the external design was retained but the center hole was provided with a pressed-in cemented carbide bushing. These last from three months to a year depending on the abrasiveness of the welding wire used. The new nozzles are now giving excellent service in about 60 machines. The guide blocks which press the wire between the feed rolls are also carbide faced.

Other Applications: Some machines have been wearproofed at as many as 72 points with cemented carbide. Beside the applications shown here, carbide can be applied to feed fingers, work support jaws, drive dogs, ratchet latches, work transfer plates, feed rolls, steadyrests, tailstock bushings, wire nippers and rolling guides, journals for dynamic balancers, arbors, work centers, collets and so on—wherever there is wear or abrasion.

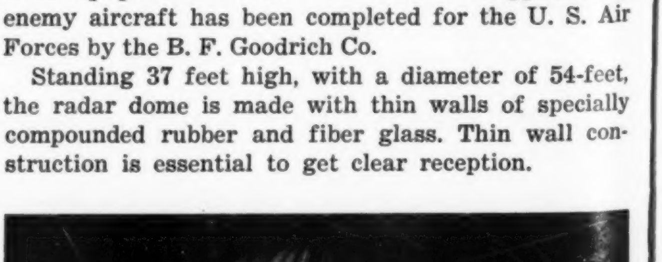
In general it will be found that about 80 per cent of the wearproofing applications permit the use of standard blanks or standard parts slightly modified. Special forms should be ordered only if a large enough quantity is to be used.

Finishing the Carbide: As far as finish goes, on carbide blanks such as guide rings the tumbled finish as supplied by the manufacturer is quite satisfactory. Other types of guides subject to high wear can be mirror-finished by polishing or lapping with diamond dust. On applications such as parts of machine mechanisms, the carbide should usually be finish-ground with a 180-grit diamond, resinoid-bonded wheel. In some cases a 320-grit wheel is desirable for finest finish. In general, the finer finishes are worth providing since there is a gain in life expectancy of the part—sometimes measurable in years.

Rubber Radar Dome

ALL-RUBBER dome to protect the delicate radar equipment needed to warn of approaching enemy aircraft has been completed for the U. S. Air Forces by the B. F. Goodrich Co.

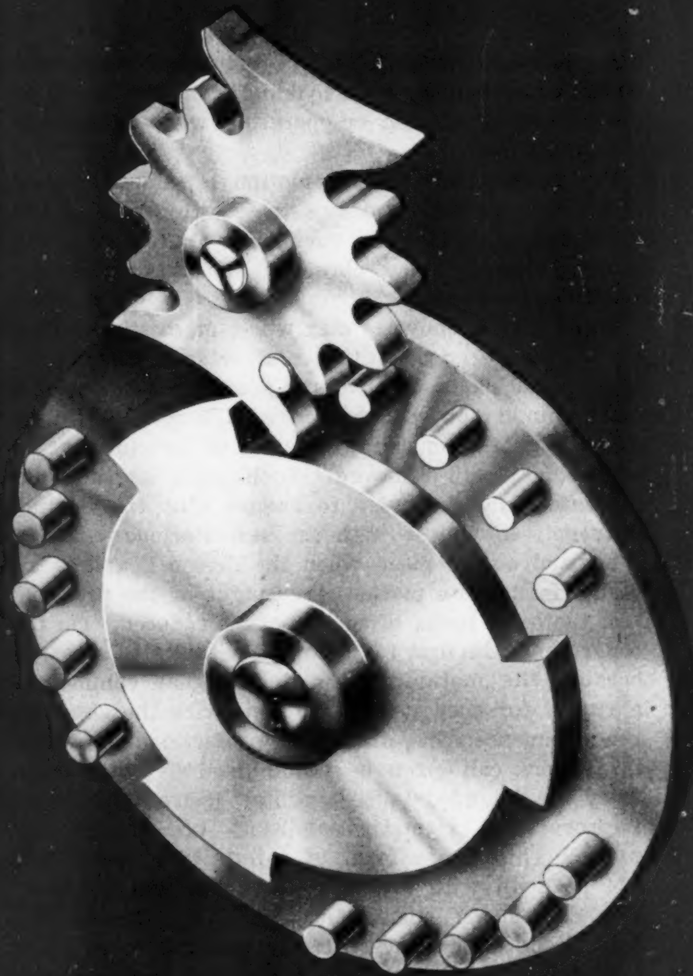
Standing 37 feet high, with a diameter of 54-feet, the radar dome is made with thin walls of specially compounded rubber and fiber glass. Thin wall construction is essential to get clear reception.



By Otto Lichtwitz
Mechanical Engineer
London, England

MECHANISMS FOR INTERMITTENT MOTION

Part 2



NUMBER of stations of a Geneva mechanism cannot be smaller than 3, or the partial movement of the driven shaft cannot exceed $\frac{1}{3}$ revolution, except in modified mechanisms similar to *Fig.* 10 (Part 1, December, 1950, MACHINE DESIGN). Yet, a frequent requirement is to impart to a shaft full revolutions with intermediate standstills. Geneva mechanisms leave little freedom because, for a given number of stations and a given center distance, all relevant dimensions and kinematic relations are determined, including the distribution of motion and standstill. Another unfavorable feature of Geneva mechanisms is that the duration of motion decreases while the partial movement of the driven gear is increasing, which accounts for the unfavorable kinematic conditions of mechanisms having a small number of stations.

External Star Wheel Mechanisms: None of the mentioned restrictions of Geneva mechanisms are imposed by star wheel mechanisms. *Figs.* 16 and 17 show mechanisms where the partial movement of the driven shaft is a full and a half revolution, respectively. *Figs.* 18 to 20 show mechanisms for partial movements of a quarter revolution. In *Fig.* 18, the motion occupies a quarter revolution of the driving gear, or the same angle as in the case of the Maltese Cross shown in *Figs.* 2 and 3. From *Figs.* 19 and 20 it is obvious that the duration of motion can be modified. In *Fig.* 19, it is increased to a full revolution of the driving gear, so that there is no standstill; in *Fig.* 20, it is reduced to less than a seventh revolution of the driving gear.

The driving gear is in all cases equipped with at least two driving rollers. The first roller serves for accelerating, the last roller for retarding, and the intermediate rollers for imparting a uniform motion to the driven gear. There is also a period of uniform motion if the driving gear has only two rollers, as in *Fig.* 20.

In *Fig.* 16, for example, the motion commences when roller 1 enters slot 1. When that roller is in the central position, the driven gear reaches maximum velocity, and retains it due to the action of rollers and teeth 2 to 7. The same velocity is still maintained when roller 8 enters into slot 8, until roller 8 reaches the central position. The driven gear is slowed down while roller 8 is receding from slot 8, the mirror image of slot 1, and comes to a standstill

when roller 8 leaves slot 8. The standstill is secured in the usual manner by a locking drum, connected to the driving gear, and co-operating with recesses in the driven gear.

Pin gearing for uniform motion is seldom applied today and, as will be demonstrated, this part of the gears can be replaced by common involute gears. Pin gearing is nevertheless chosen in the illustrations for clarity and simplicity.

In *Fig.* 31 is shown a star wheel mechanism where the means for accelerating and retarding are combined with parts of involute gears. This arrangement should not be confused with a form of intermittent gears consisting merely of involute gears with some teeth removed. Such gears are actually mutilated, and in order to ensure that the driving gear gets into mesh with the still stationary driven gear, and out of mesh when the driven gear has already finished its motion, the height of the first and last tooth, or teeth, of such gears is reduced. On the other hand, the first tooth should actually be stronger because the motion of the driven gear commences abruptly for such gears. To cope with the shock at the commencement of motion, projections of doubtful shapes can often be found fixed to both gears. They are intended to be auxiliary gears, but neither improve the kinematic conditions nor remove the noise. This crude form of intermittent gears will not be dealt with for spur gears, for which a more perfect solution exists, but it will be discussed in a later issue under intermittently working bevel gears.

If the elements for uniform motion are separated from those for nonuniform motion, as in *Fig.* 31, more than one revolution can be imparted to the driven shaft during a partial movement of the driving shaft. However, since the driving gear in such cases becomes rather bulky, and since a multiplication of the movement of the driven shaft can be achieved by simple means, such cases will not be considered here.

GEOMETRY: In *Fig.* 21*a* a star wheel mechanism is shown in the position where the driven gear commences moving. The shape of the accelerating slot is determined by the requirement that the roller must recede from the slot when, subsequently, both gears are rotating with uniform velocity. The center line of the slot must, therefore, be a part of an epicycloid which is traced by a point on the circumference of the pitch circle 1 of the driving gear when rolling without sliding on the exterior of the pitch circle 2 of the driven gear. For the start and finish of the motion without shock, the epicycloid and the pitch circle 1 must have a common tangent at the point *D*.

A circle of the size of the pitch circle of the driving gear may be imagined touching the pitch circle of the driven gear at the point *C*, and rolling afterwards along that circle until it gets into the position 1′ for which the point of contact is at *E*, and the generating point of the epicycloid at *D*. The center of the rolling circle is then at *A*′.

As the triangles *ABD* and *A′BD* are congruent (equal sides) the angles *DA′B* and *DAB* are equal, namely the angle α_0 through which the driving roller has to rotate from the initial to the central position. If the line *AE* is extended to the further intersection *F* with the circle 2, the resulting isosceles tri-

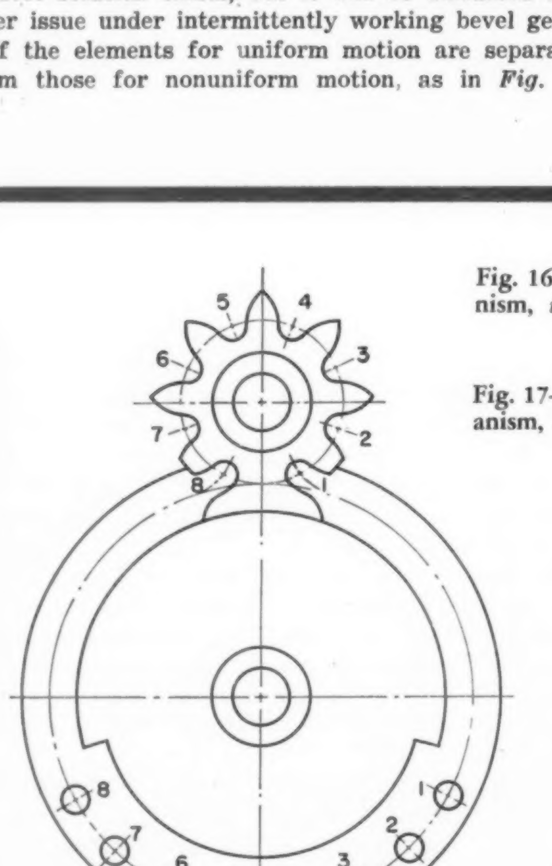
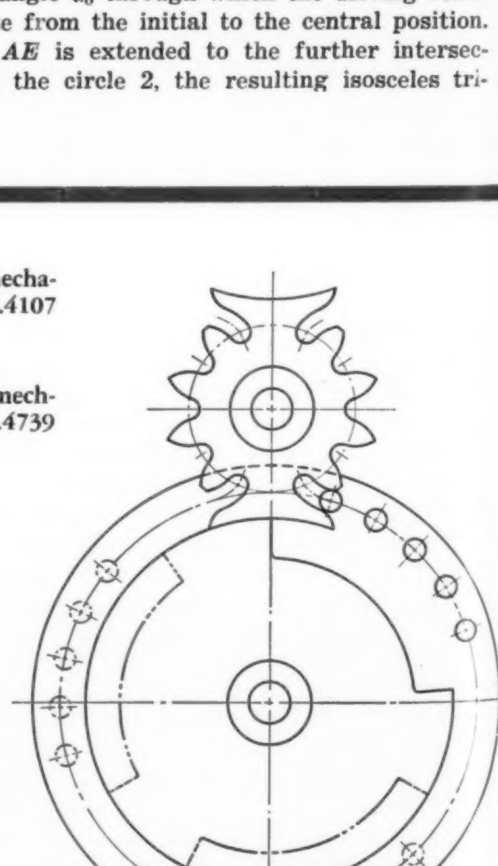


Fig. 16—Left—Star wheel mechanism, $n=1$, $\mu=0.375$, $\epsilon=0.4107$

Fig. 17—Right—Star wheel mechanism, $n=2$, $\mu=0.400$, $\epsilon=0.4739$



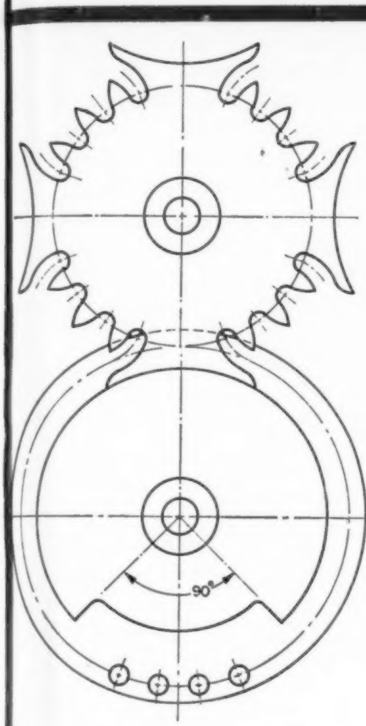


Fig. 18—Star wheel mechanism, $n=4$, $\mu=0.7773$, $\epsilon=1$

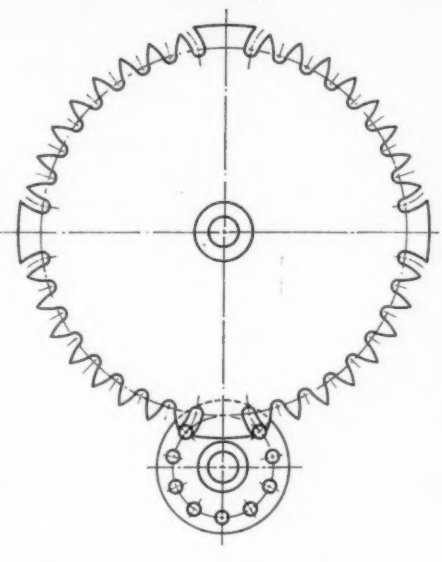


Fig. 19—Star wheel mechanism, $n=4$, $\mu=3.6233$, $\epsilon=\epsilon_{max}=4$

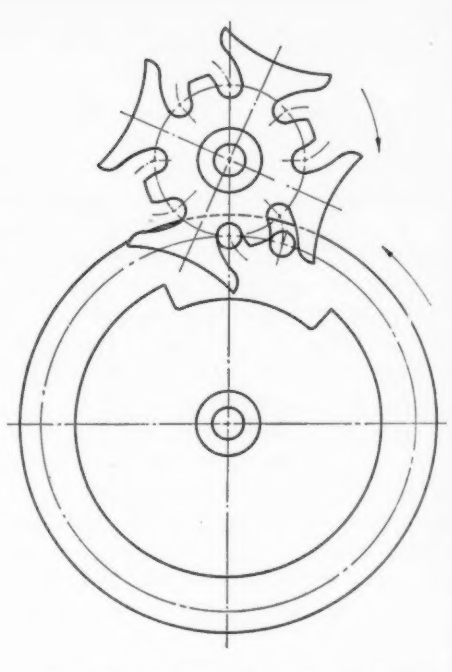


Fig. 20—Star wheel mechanism, $n=4$, $\mu=0.3943$, $\epsilon=\epsilon_{min}=0.5419$

angle *EBF* is similar to the triangle *EA'D*, so that the angle *EBF* is likewise α_0. A further triangle similar to the previous is the isosceles triangle *BAF*, two sides of which are equal to the sum of the radii of the two gears, $r_2 + r_1$, and the third side of which is the radius r_2 of the driven gear. It follows that $\sin \alpha_0/2 = r_2/2(r_2+r_1)$, or with $\mu = r_2/r_1$,

$$\sin \frac{\alpha_0}{2} = \frac{\mu}{2(1+\mu)} \quad \text{(20)}$$

$$\alpha_0 = 2 \sin^{-1} \frac{\mu}{2(1+\mu)} \quad \text{(21)}$$

The line *BG* which bisects the angle *EBF* is perpendicular to the line *AEFH*, and the line *BH* is perpendicular to *AB*. The angle *GBH* is therefore α_0, and the angle *FBH* is $\frac{1}{2}\alpha_0$. Since the angle *EBH* is equal to $\alpha_0 + \frac{1}{2}\alpha_0 = 1\frac{1}{2}\alpha_0$, the angle *C'BE* is $\frac{1}{2}\pi - 1\frac{1}{2}\alpha_0$, and the angle ϕ_0 which determines the outer end of the slot is half of it, or

$$\phi_0 = \frac{\pi}{4} - \frac{3\alpha_0}{4} = \frac{\pi}{4} - \frac{3}{2} \sin^{-1} \frac{\mu}{2(1+\mu)} \quad \text{(22)}$$

The arcs *CE* and *DE* are equal, and the latter is $r_1\alpha_0$. The angle *CBE* is, therefore, $r_1\alpha_0/r_2 = \alpha_0/\mu$, and it follows from *Fig. 21a* that the angle β_0, which determines the inner end of the slot, is

$$\beta_0 = \frac{\pi}{2} - \frac{\alpha_0}{\mu} - \frac{3\alpha_0}{2} = \frac{\pi}{2} - \frac{\alpha_0}{2}\left(\frac{2+3\mu}{\mu}\right)$$

$$= \frac{\pi}{2} - \left(\frac{2+3\mu}{\mu}\right) \sin^{-1} \frac{\mu}{2(1+\mu)} \quad \text{(23)}$$

If the center distance of driving and driven gear is $a = r_2 + r_1$, the radii of the two gears are

$$r_1 = \frac{a}{1+\mu} \quad \text{(24)}$$

$$r_2 = \frac{a\mu}{1+\mu} \quad \text{(25)}$$

The distance $s = DC'$ is

$$s = 2 r_1 \sin \frac{\alpha_0}{2} = \frac{r_1 r_2}{r_1 + r_2} = \frac{a\mu}{(1+\mu)^2} \quad \text{(26)}$$

The point *D* can be constructed by means of the calculated value of *s*, or entirely graphically in the following way: a circle with the center at *A*, and the radius *AB* is scribed; the point of intersection *F* with the pitch circle of the driven gear is connected with the point *A*; that line intersects the pitch circle of the driving gear at the required point *D*, as it can easily be proved that the distance *C'D* complies with the value found in Equation 26.

The distance $\rho_0 = BD$ can be obtained from the triangle *C'BD*, in which the angle $BC'D = \frac{1}{2}\pi + \frac{1}{2}\alpha_0$, and the angle $C'BD = \phi_0 = \frac{1}{4}\pi - \frac{3}{4}\alpha_0$. The third angle $C'DB = \pi - (\frac{1}{2}\pi + \frac{1}{2}\alpha_0) - (\frac{1}{4}\pi - \frac{3}{4}\alpha_0) = \frac{1}{4}\pi + \frac{1}{4}\alpha_0$. Using the sine rule, we obtain $\rho_0/r_2 = \sin(\frac{1}{2}\pi + \frac{1}{2}\alpha_0) \div \sin(\frac{1}{4}\pi + \frac{1}{4}\alpha_0) = 2\cos(\frac{1}{4}\pi + \frac{1}{4}\alpha_0)$, and it follows that $\rho_0 = 2r_2\cos(\frac{1}{4}\pi + \frac{1}{4}\alpha_0)$, or that

$$\rho_0 = 2\,a\,\frac{\mu}{1+\mu}\cos\left(\frac{\pi}{4} + \frac{\alpha_0}{4}\right) \quad \text{(27)}$$

If the driven gear has *n* equally spaced locking shoes, one partial movement comprises the angle

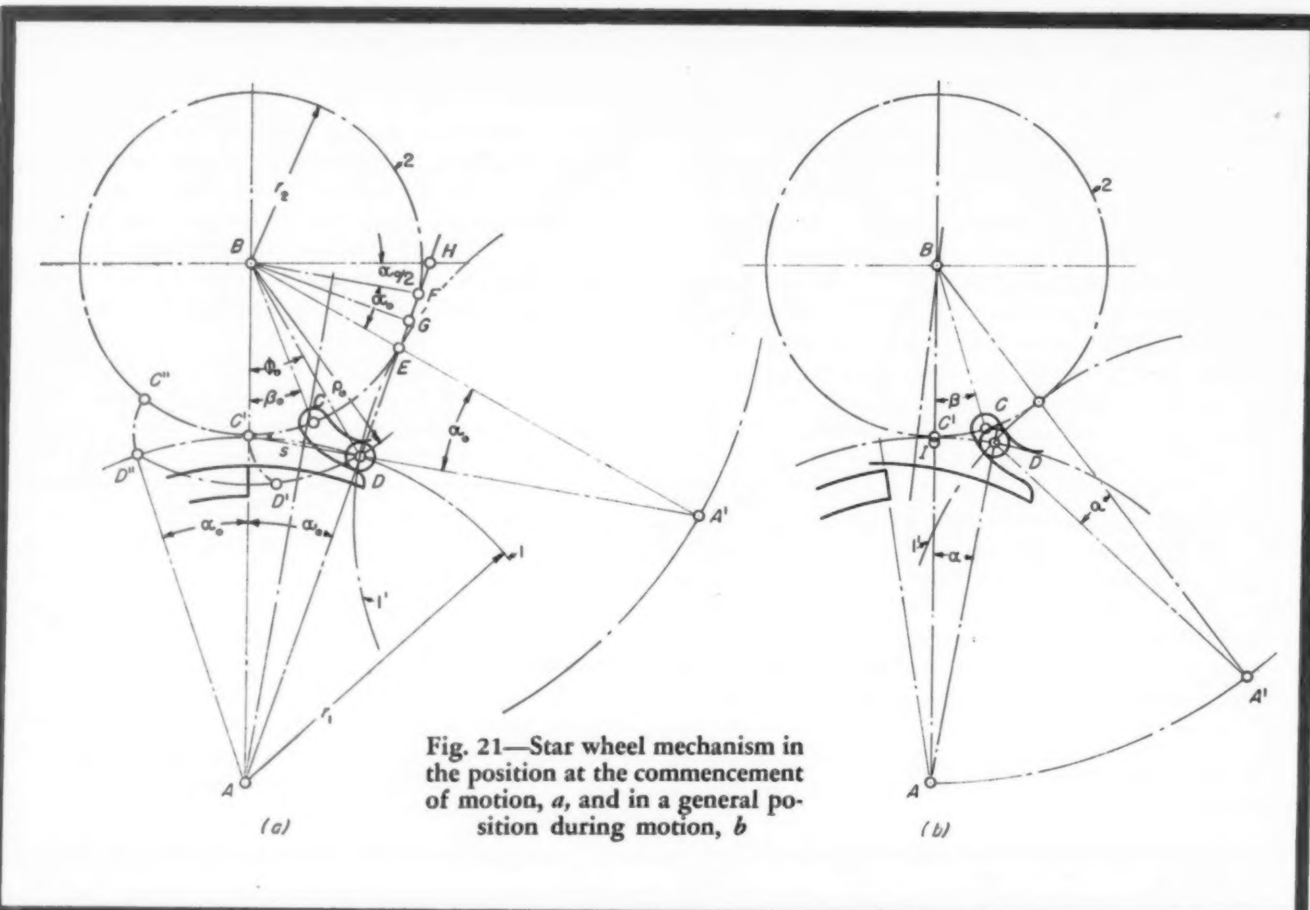


Fig. 21—Star wheel mechanism in the position at the commencement of motion, *a*, and in a general position during motion, *b*

$2\pi/n$. During the angle α_0 of the driving gear, the driven gear is accelerated, and rotates through the angle β_0. During another angle α_0 of the driving gear, the driven gear is retarded and rotates likewise through the angle β_0. The difference $(2\pi/n) - 2\beta_0$ is the angle through which the driven gear rotates with constant angular velocity, and the driving gear rotates in that period through angle $\mu[(2\pi/n) - 2\beta_0]$.

One partial movement of the driven gear thus requires the angle $2\alpha_0 + \mu\,[(2\pi/n) - 2\beta_0]$ of the driving gear, and the ratio of gearing is

$$\varepsilon = \frac{2\alpha_0 + \mu\left(\dfrac{2\pi}{n} - 2\beta_0\right)}{\dfrac{2\pi}{n}}$$

or, from Equations 21 and 23,

$$\varepsilon = \mu + n\,\frac{4 + 3\mu}{\pi}\sin^{-1}\frac{\mu}{2(1+\mu)} - \frac{n\mu}{2} \quad \ldots\ldots (28)$$

In the frequent special case of $n = 1$,

$$\varepsilon_1 = \frac{\mu}{2} + \frac{4 + 3\mu}{\pi}\sin^{-1}\frac{\mu}{2(1+\mu)} \quad \ldots\ldots\ldots\ldots (29)$$

Similarly, as with Geneva mechanisms, ϵ and μ are not identical; ϵ is always larger than μ. Either ϵ or μ can be arbitrarily chosen within certain limits.

The upper limit of ϵ is reached if the duration of the uniform motion is increased to such an extent that the standstill is reduced to zero, so that $2\alpha_0 + \mu \times [(2\pi/n) - 2\beta_0] = 2\pi$. In this case, $\epsilon = 2\pi/(2\pi/n) = n$, and it follows that

$$\varepsilon_{max} = n \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (30)$$

The value of μ which pertains to the maximum of ϵ is obtained by combining Equations 28 and 30. That is,

$$\mu_{max} + n\,\frac{4 + 3\mu_{max}}{\pi}\sin^{-1}\frac{\mu_{max}}{2(1+\mu_{max})} - \frac{n\,\mu_{max}}{2} = n$$

or

$$\frac{4 + 3\,\mu_{max}}{\pi\,\mu_{max}}\sin^{-1}\frac{\mu_{max}}{2(1+\mu_{max})} - \frac{1}{\mu_{max}} = \frac{n-2}{2\,n} \quad \ldots (31)$$

The lower limit of ϵ pertains to the smallest possible angle during which the driven gear rotates with

Table 4—Extreme Values for External Star Wheel Mechanisms

n Stations	ε_{max} Eq. 30	μ_{max} Eq. 31	ε_{min} Eq. 32	μ_{min} Eq. 33	m_{max} Eq. 49
1	1	0.9388	0	0	∞
2	2	1.8406	0	0	∞
3	3	2.7336	0.03373	0.02411	88
4	4	3.6233	0.5419	0.3943	7
5	5	4.5114	1.0467	0.7696	4
6	6	5.3983	1.5486	1.1483	3
7	7	6.2840	2.0518	1.5280	3
8	8	7.1709	2.5482	1.9092	3
9	9	8.0569	3.0555	2.2911	2
10	10	8.9430	3.5565	2.6731	2

constant velocity. That period cannot be eliminated altogether. In *Fig. 21a*, *CD* is the position of the slot at the commencement of motion, and *C'D'* is the position when the driven gear reaches the full angular velocity. That velocity must be maintained at least until the accelerating roller leaves the slot at *D''*, when the slot occupies the position *C''D''*.

The driving gear rotates, from the position *D* to *D''* of the driving roller through the angle $2\alpha_0$, and the retardation requires at least another period of α_0, so that the condition for the minimum of ϵ is

$$\epsilon_{min} = \frac{3\,\alpha_{0-min}}{\dfrac{2\,\pi}{n}} = \frac{3\,n}{\pi}\,\frac{\alpha_{0-min}}{2}$$

$$= \frac{3\,n}{\pi}\,\sin^{-1}\frac{\mu_{min}}{2(1+\mu_{min})} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots (32)$$

By combining Equations 28 and 32, the condition for the minimum of μ is found in

$$\mu_{min} + n\,\frac{4+3\,\mu_{min}}{\pi}\,\sin^{-1}\frac{\mu_{min}}{2(1+\mu_{min})} - \frac{n\,\mu_{min}}{2} = \frac{3\,n}{\pi}\,\sin^{-1}\frac{\mu_{min}}{2(1+\mu_{min})}$$

or

$$\frac{1+3\,\mu_{min}}{\mu_{min}}\,\sin^{-1}\frac{\mu_{min}}{2(1+\mu_{min})} = \frac{\pi}{2}\,\frac{n-2}{n} \quad \ldots\ldots (33)$$

By combining Equations 32 and 33, it is found that

$$\epsilon_{min} = \frac{3(n-2)\mu_{min}}{2(1+3\,\mu_{min})} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots (34)$$

That relation between ϵ_{min} and μ_{min} is more convenient than Equation 32, since it does not contain arcs.

The condition for values of μ above the minimum is analogously

$$\frac{1+3\,\mu}{\mu}\,\sin^{-1}\frac{\mu}{2(1+\mu)} > \frac{\pi}{2}\,\frac{n-2}{n}$$

For $\mu = 0$, that inequality assumes the form $\frac{1}{2} > \frac{1}{2}\pi(n-2)/n$, or $n < 2\pi/(\pi-1) = 2.93$. That indicates, for $n = 1$ or 2, μ can be kept as small as desired, and even zero. Star wheel mechanisms with $\mu = 0$ are shown in *Figs.* 22 and 23.

To find the extreme values of μ for a given number of stations, the transcendent equations of Equations 31 and 33 must be solved, but TABLE 4 contains the solutions for the most usual numbers of stations. With four stations, for example, TABLE 4 indicates the maximum of $\epsilon = 4$, with the corresponding $\mu = 3.623$ (*Fig.* 19). The minimum of ϵ is 0.5419, with the corresponding $\mu = 0.3943$ (*Fig.* 20).

The values of μ and ϵ increase with increasing number of stations; *Figs.* 24 and 25 show mechanisms where n, μ, and ϵ, are infinite. While the rack-like gears in *Figs.* 22 and 23 are the driving members, they are the driven ones in *Figs.* 24 and 25.

Although the investigations of geometrical and kinematic properties are best based on the ratio of the pitch radii (μ), the movement of an intermittently rotating shaft is usually specified by the ratio of the angles of driving and driven gears (ϵ). In such cases, μ must be determined by Equation 28, but here again TABLE 5 renders the solution of transcendent equations unnecessary as it contains the ratios ϵ likely to be met in practice.

The relation between ϵ and μ, expressed by Equation 28, is of a nature which renders ϵ an odd value if a round figure is chosen for μ, and vice versa. Round figures of μ are advantageous because the layout of the elements for imparting the uniform mo-

Table 5—Relation Between Basic Properties for External Star Wheel Mechanisms

	μ									
ϵ	$n = 1$	$n = 2$	$n = 3$	$n = 4$	$n = 5$	$n = 6$	$n = 7$	$n = 8$	$n = 9$	$n = 10$
1/8=0.12500	0.11152	0.10043	0.09121	.....	.....	.....	.....	.....	.....	.....
1/7=0.14286	0.12767	0.11512	0.10464	.....	.....	.....	.....	.....	.....	.....
1/6=0.16667	0.14930	0.13483	0.12268	.....	.....	.....	.....	.....	.....	.....
1/5=0.2000	0.17970	0.16264	0.14821	.....	.....	.....	.....	.....	.....	.....
1/4=0.2500	0.2256	0.2048	0.18706	.....	.....	.....	.....	.....	.....	.....
1/3=0.3333	0.3027	0.2762	0.2532	.....	.....	.....	.....	.....	.....	.....
3/8=0.3750	0.3416	0.3124	0.2869	.....	.....	.....	.....	.....	.....	.....
2/5=0.4000	0.3650	0.3343	0.3073	.....	.....	.....	.....	.....	.....	.....
1/2=0.5000	0.4591	0.4226	0.3901	.....	.....	.....	.....	.....	.....	.....
3/5=0.6000	0.5540	0.5122	0.4747	0.4407	.....	.....	.....	.....	.....	.....
5/8=0.6250	0.5778	0.5350	0.4962	0.4611	.....	.....	.....	.....	.....	.....
2/3=0.6667	0.6176	0.5728	0.5322	0.4952	.....	.....	.....	.....	.....	.....
3/4=0.7500	0.6975	0.6491	0.6047	0.5641	.....	.....	.....	.....	.....	.....
4/5=0.8000	0.7455	0.6951	0.6487	0.6060	.....	.....	.....	.....	.....	.....
5/6=0.8333	0.7776	0.7259	0.6782	0.6341	.....	.....	.....	.....	.....	.....
1=1.0000	0.9388	0.8813	0.8275	0.7773	.....	.....	.....	.....	.....	.....
1-1/2=1.5000	.....	1.3567	1.2892	1.2248	1.1633	.....	.....	.....	.....	.....
2=2.0000	.....	1.8406	1.7641	1.6901	1.6185	1.5484	.....	.....	.....	.....
2-1/2=2.5000	.....	.....	2.2466	2.1654	2.0863	2.0090	1.9334	.....	.....	.....
3=3.0000	.....	.....	2.7336	2.6477	2.5633	2.4800	2.3984	2.3185	.....	.....
3-1/2=3.5000	.....	.....	.....	3.1341	3.0470	2.9577	2.8715	2.7866	2.7034	.....
4=4.0000	.....	.....	.....	3.6233	3.5317	3.4392	3.3504	3.2620	3.1741	3.0878
4-1/2=4.5000	.....	.....	.....	.....	4.0200	3.9267	3.8342	3.7418	3.6513	3.5609
5=5.0000	.....	.....	.....	.....	4.5114	4.4150	4.3203	4.2261	4.1329	4.0400
6=6.0000	.....	.....	.....	.....	.....	5.3983	5.2994	5.2009	5.1000	5.0069
7=7.0000	.....	.....	.....	.....	.....	.....	6.2840	6.1837	6.0834	5.9850
8=8.0000	.....	.....	.....	.....	.....	.....	.....	7.1709	7.0675	6.9663
9=9.0000	.....	.....	.....	.....	.....	.....	.....	.....	8.0569	7.9520
10=10.0000	.....	.....	.....	.....	.....	.....	.....	.....	.....	8.9430

Table 6—Properties of External Star Wheel Mechanisms

μ	α_0	ϕ_0	β_0	$\frac{r_1}{a}$	$\frac{r_2}{a}$	$\frac{s}{a}$	$\frac{\rho_0}{a}$	ε	ν	γ	n for ε, ν & γ	$\left(\frac{d\beta}{d\alpha}\right)_0$	$\left(\frac{d^2\beta}{d\alpha^2}\right)_{-0}$	α_{max}	$\left(\frac{d^2\beta}{d\alpha^2}\right)_{max}$
	Eq. 21	Eq. 22	Eq. 23	Eq. 24	Eq. 25	Eq. 26	Eq. 27	Eq. 28	Eq. 35	Eq. 39		Eq. 43	Eq. 45	Eq. 47	Eq. 48
0.00	0°	45°	32°42′	1	0	0	0	0	0	360°	1	∞	∞	0°	∞
0.02	1° 7′	44° 9′	32° 9′	0.9804	0.01961	0.01922	0.02759	0.02267	0.02267	351°50′	1	50.00	2575	0°38′	3338
0.04	2°12′	43°21′	31°36′	0.9615	0.03846	0.03698	0.05387	0.04522	0.04522	343°47′	1	25.00	662.6	1°16′	834.8
0.06	3°15′	42°34′	31° 5′	0.9434	0.05660	0.05340	0.07891	0.06766	0.06766	335°39′	1	16.67	302.9	1°54′	382.5
0.08	4°15′	41°49′	30°34′	0.9259	0.07407	0.06859	0.10280	0.0900	0.0900	327°36′	1	12.50	175.1	2°32′	219.1
0.10	5°13′	41° 6′	30° 5′	0.9091	0.09091	0.08264	0.12561	0.11224	0.11224	319°36′	1	10.00	115.1	3° 9′	142.5
0.12	6° 9′	40°24′	29°36′	0.8929	0.10714	0.09566	0.14741	0.13438	0.13438	311°37′	1	8.33	82.06	3°45′	101.3
0.14	7° 2′	39°43′	29° 9′	0.8772	0.12281	0.10773	0.16827	0.15645	0.15645	303°41′	1	7.14	61.85	4°20′	75.65
0.16	7°55′	39° 4′	28°42′	0.8621	0.13793	0.11891	0.18822	0.17843	0.17843	295°46′	1	6.25	48.55	4°54′	58.93
0.18	8°45′	38°27′	28°17′	0.8475	0.15254	0.12927	0.2074	0.2003	0.2003	287°53′	1	5.56	39.31	5°28′	47.41
0.20	9°34′	37°50′	27°52′	0.8333	0.16667	0.13889	0.2257	0.2222	0.2222	280° 1′	1	5.00	32.61	6° 2′	39.10
0.22	10°21′	37°14′	27°27′	0.8197	0.18033	0.14781	0.2432	0.2439	0.2439	272°11′	1	4.54	27.59	6°35′	32.86
0.24	11° 6′	36°40′	27° 4′	0.8065	0.19355	0.15609	0.2601	0.2656	0.2656	264°23′	1	4.17	23.72	7° 7′	28.06
0.26	11°51′	36° 7′	26°41′	0.7937	0.2063	0.16377	0.2764	0.2873	0.2873	256°35′	1	3.85	20.67	7°39′	24.32
0.28	12°34′	35°35′	26°19′	0.7813	0.2187	0.17090	0.2920	0.3088	0.3088	248°49′	1	3.57	18.22	8°10′	21.32
0.30	13°15′	35° 4′	25°57′	0.7692	0.2308	0.17752	0.3070	0.3304	0.3304	241° 4′	1	3.33	16.23	8°41′	18.87
0.32	13°55′	34°33′	25°36′	0.7576	0.2424	0.18793	0.3214	0.3518	0.3518	233°20′	1	3.13	14.56	9°11′	16.85
0.34	14°35′	34° 4′	25°16′	0.7463	0.2537	0.18936	0.3353	0.3733	0.3733	225°37′	1	2.94	13.17	9°41′	15.16
0.36	15°13′	33°36′	24°56′	0.7353	0.2647	0.19463	0.3487	0.3946	0.3946	217°56′	1	2.78	11.99	10°10′	13.74
0.38	15°50′	33° 8′	24°37′	0.7246	0.2754	0.19954	0.3616	0.4160	0.4160	210°15′	1	2.63	10.98	10°39′	12.52
0.40	16°26′	32°41′	24°18′	0.7143	0.2857	0.2041	0.3740	0.4373	0.4373	202°35′	1	2.50	10.10	11° 8′	11.47
0.42	17° 1′	32°14′	23°59′	0.7042	0.2958	0.2083	0.3861	0.4585	0.4585	194°56′	1	2.38	9.343	11°36′	10.57
0.44	17°35′	31°49′	23°41′	0.6944	0.3056	0.2122	0.3978	0.4797	0.4797	187°18′	1	2.27	8.676	12° 4′	9.767
0.46	18° 8′	31°24′	23°24′	0.6849	0.3151	0.2158	0.4090	0.5009	0.5009	179°40′	1	2.17	8.088	12°31′	9.069
0.48	18°48′	31° 0′	23° 7′	0.6757	0.3243	0.2191	0.4198	0.5221	0.5221	172° 4′	1	2.08	7.565	12°58′	8.450
0.50	19°11′	30°36′	22°50′	0.6667	0.3333	0.2222	0.4303	0.5432	0.5432	164°28′	1	2.00	7.099	13°25′	7.900
0.52	19°42′	30°13′	22°34′	0.6579	0.3421	0.2251	0.4405	0.5642	0.5642	156°53′	1	1.92	6.681	13°52′	7.400
0.54	20°12′	29°51′	22°19′	0.6494	0.3506	0.2277	0.4504	0.5853	0.5853	149°18′	1	1.85	6.305	14°18′	6.966
0.56	20°41′	29°29′	22° 3′	0.6410	0.3590	0.2301	0.4599	0.6063	0.6063	141°44′	1	1.79	5.964	14°43′	6.567
0.58	21° 9′	29° 8′	21°48′	0.6329	0.3671	0.2323	0.4691	0.6273	0.6273	134°11′	1	1.72	5.655	15° 8′	6.207
0.60	21°37′	28°47′	21°33′	0.6250	0.3750	0.2344	0.4780	0.6482	0.6482	126°38′	1	1.67	5.373	15°33′	5.879
0.62	22° 4′	28°27′	21°19′	0.6173	0.3827	0.2362	0.4867	0.6692	0.6692	119° 6′	1	1.61	5.115	15°57′	5.580
0.64	22°30′	28° 7′	21° 5′	0.6098	0.3902	0.2380	0.4951	0.6901	0.6901	111°35′	1	1.56	4.879	16°21′	5.306
0.66	22°56′	27°48′	20°52′	0.6024	0.3976	0.2395	0.5033	0.7109	0.7109	104° 5′	1	1.52	4.662	16°45′	5.055
0.68	23°21′	27°29′	20°38′	0.5952	0.4048	0.2409	0.5113	0.7318	0.7318	96°33′	1	1.47	4.461	17° 9′	4.824
0.70	23°46′	27°11′	20°25′	0.5882	0.4118	0.2422	0.5190	0.7526	0.7526	89° 3′	1	1.43	4.275	17°33′	4.611
0.72	24°10′	26°53′	20°12′	0.5814	0.4186	0.2434	0.5265	0.7734	0.7734	81°34′	1	1.39	4.103	17°56′	4.414
0.74	24°33′	26°35′	19°59′	0.5747	0.4253	0.2444	0.5337	0.7942	0.7942	74° 4′	1	1.35	3.943	18°19′	4.231
0.76	24°56′	26°18′	19°47′	0.5682	0.4318	0.2453	0.5407	0.8150	0.8150	66°35′	1	1.32	3.794	18°42′	4.062
0.78	25°19′	26° 1′	19°35′	0.5618	0.4382	0.2462	0.5476	0.8358	0.8358	59° 7′	1	1.28	3.656	19° 4′	3.904
0.80	25°41′	25°44′	19°23′	0.5556	0.4444	0.2469	0.5543	0.8565	0.8565	51°39′	1	1.25	3.526	19°26′	3.757
0.82	26° 2′	25°28′	19°11′	0.5495	0.4505	0.2476	0.5608	0.8772	0.8772	44°12′	1	1.22	3.404	19°48′	3.619
0.84	26°23′	25°12′	19° 0′	0.5435	0.4565	0.2481	0.5671	0.8979	0.8979	36°44′	1	1.19	3.290	20° 9′	3.491
0.86	26°44′	24°57′	18°49′	0.5376	0.4624	0.2486	0.5733	0.9186	0.9186	29°17′	1	1.16	3.182	20°30′	3.370
0.88	27° 4′	24°42′	18°38′	0.5319	0.4681	0.2490	0.5794	0.9393	0.9393	21°51′	1	1.14	3.081	20°52′	3.256
0.90	27°24′	24°27′	18°28′	0.5263	0.4737	0.2493	0.5852	0.9600	0.9600	14°25′	1	1.11	2.986	21°13′	3.149
0.92	27°43′	24°12′	18°17′	0.5208	0.4792	0.2496	0.5909	0.9806	0.9806	6°59′	1	1.09	2.896	21°34′	3.049
0.94	28° 2′	23°58′	18° 6′	0.5155	0.4845	0.2498	0.5964	1.0625	0.5312	168°46′	2	1.06	2.811	21°54′	2.954
0.96	28°21′	23°44′	17°56′	0.5102	0.4898	0.2499	0.6019	1.0837	0.5418	164°56′	2	1.04	2.730	22°14′	2.864
0.98	28°39′	23°30′	17°46′	0.5051	0.4949	0.2500	0.6071	1.1049	0.5524	161° 8′	2	1.02	2.654	22°34′	2.779
1.00	28°57′	23°17′	17°37′	0.5000	0.5000	0.2500	[illegible]	[illegible]	0.5630	157°19′	2	[illegible]	[illegible]	22°54′	[illegible]

Table 6—Properties of External Star Wheel Mechanisms

(Continued)

μ	α_0	ϕ_0	β_0	$\frac{r_1}{a}$	$\frac{r_2}{a}$	$\frac{s}{a}$	$\frac{\rho_0}{a}$	ε	ν	γ	n for ε, ν & γ	$\left(\frac{d\beta}{d\alpha}\right)_0$	$\left(\frac{d^2\beta}{d\alpha^2}\right)_{-\alpha 0}$	α_{max}	$\left(\frac{d^2\beta}{d\alpha^2}\right)_{max}$
	Eq. 21	Eq. 22	Eq. 23	Eq. 24	Eq. 25	Eq. 26	Eq. 27	Eq. 28	Eq. 35	Eq. 39		Eq. 43	Eq. 45	Eq. 47	Eq. 48
1.1	30°22′	22°14′	16°51′	0.4762	0.5238	0.2494	0.6364	1.2315	0.6158	138°19′	2	0.909	2.270	24°29′	2.355
1.2	31°39′	21°16′	16° 9′	0.4545	0.5455	0.2479	0.6578	1.3365	0.6682	119°26′	2	0.833	2.021	26° 2′	2.084
1.3	32°50′	20°23′	15°30′	0.4348	0.5625	0.2457	0.6770	1.4410	0.7205	100°38′	2	0.769	1.820	27°30′	1.866
1.4	33°55′	19°34′	14°54′	0.4167	0.5833	0.2431	0.6943	1.5451	0.7725	81°53′	2	0.714	1.654	28°53′	1.688
1.5	34°55′	18°49′	14°21′	0.4000	0.6000	0.2400	0.7099	1.6488	0.8244	63°13′	2	0.667	1.514	30°12′	1.540
1.6	35°50′	18° 7′	13°50′	0.3846	0.6154	0.2367	0.7241	1.7522	0.8761	44°36′	2	0.625	1.395	31°28′	1.414
1.7	36°42′	17°29′	13°22′	0.3704	0.6296	0.2332	0.7371	1.8554	0.9277	26° 2′	2	0.588	1.294	32°42′	1.308
1.8	37°30′	16°53′	12°55′	0.3571	0.6429	0.2296	0.7489	1.9583	0.9792	7°31′	2	0.556	1.206	33°52′	1.216
1.9	38°15′	16°19′	12°30′	0.3448	0.6552	0.2259	0.7598	2.1414	0.7138	103° 2′	3	0.526	1.129	35° 0′	1.136
2.0	38°57′	15°48′	12° 7′	0.3333	0.6667	0.2222	0.7698	2.2453	0.7484	90°34′	3	0.500	1.061	36° 5′	1.065
2.2	40°13′	14°50′	11°24′	0.3125	0.6875	0.2148	0.7875	2.4520	0.8173	65°46′	3	0.455	0.9461	38° 8′	0.9481
2.4	41°20′	14° 0′	10°47′	0.2941	0.7059	0.2076	0.8030	2.6579	0.8860	31° 3′	3	0.417	0.8536	40° 3′	0.8542
2.6	42°20′	13°15′	10°13′	0.2778	0.7222	0.2006	0.8164	2.8628	0.9543	6°26′	3	0.385	0.7773	41°50′	0.7774
2.8	43°14′	12°34′	9°42′	0.2632	0.7368	0.19391	0.8281	3.1571	0.7893	75°52′	4	0.357	0.7134	.....	.....
3.0	44°03′	11°58′	9°15′	0.2500	0.7500	0.18750	0.8386	3.3626	0.8406	57°22′	4	0.333	0.6592	.....	.....
3.2	44°47′	11°25′	8°50′	0.2381	0.7619	0.18140	0.8478	3.5675	0.8919	38°56′	4	0.313	0.6126	.....	.....
3.4	45°27′	10°55′	8°27′	0.2273	0.7727	0.17562	0.8561	3.7722	0.9430	20°31′	4	0.294	0.5721	.....	.....
3.6	46° 4′	10°27′	8° 6′	0.2174	0.7826	0.17013	0.8635	3.9763	0.9941	2° 8′	4	0.278	0.5366	.....	.....
3.8	46°38′	10° 2′	7°47′	0.2083	0.7917	0.16493	0.8702	4.2750	0.8550	52°13′	5	0.263	0.5052	.....	.....
4.0	47° 9′	9°38′	7°29′	0.2000	0.8000	0.16000	0.8764	4.4792	0.8958	37°30′	5	0.250	0.4774	.....	.....
4.2	47°38′	9°16′	7°12′	0.19231	0.8077	0.15533	0.8819	4.6832	0.9366	22°49′	5	0.238	0.4524	.....	.....
4.4	48° 5′	8°56′	6°57′	0.18519	0.8148	0.15089	0.8870	4.8868	0.9774	8° 9′	5	0.227	0.4298	.....	.....
4.6	48°30′	8°37′	6°42′	0.17857	0.8214	0.14668	0.8917	5.1883	0.8647	48°42′	6	0.217	0.4094	.....	.....
4.8	48°53′	8°20′	6°29′	0.17241	0.8276	0.14269	0.8961	5.3921	0.8987	36°30′	6	0.208	0.3909	.....	.....
5.0	49°15′	8° 4′	6°17′	0.16667	0.8333	0.13889	0.9001	5.5952	0.9326	24°17′	6	0.200	0.3740	.....	.....
5.2	49°35′	7°49′	6° 5′	0.16129	0.8387	0.13528	0.9038	5.7985	0.9664	12° 5′	6	0.192	0.3585	.....	.....
5.4	49°54′	7°34′	5°54′	0.15625	0.8437	0.13183	0.9072	6.1018	0.8719	46°12′	7	0.185	0.3442	.....	.....
5.6	50°12′	7°21′	5°44′	0.15152	0.8485	0.12856	0.9105	6.3057	0.9008	35°42′	7	0.179	0.3310	.....	.....
5.8	50°29′	7° 8′	5°34′	0.14706	0.8529	0.12543	0.9135	6.5086	0.9297	25°17′	7	0.172	0.3188	.....	.....
6.0	50°45′	6°56′	5°24′	0.14286	0.8571	0.12245	0.9162	6.7110	0.9588	14°52′	7	0.167	0.3075	.....	.....
6.2	51° 0′	6°45′	5°16′	0.13889	0.8611	0.12239	0.9190	6.9145	0.9878	4°24′	7	0.161	0.2969	.....	.....
6.4	51°15′	6°34′	5° 8′	0.13514	0.8649	0.11688	0.9215	7.2200	0.9025	35° 7′	8	0.156	0.2870	.....	.....
6.6	51°28′	6°24′	5° 0′	0.13158	0.8684	0.11427	0.9238	7.4225	0.9278	25°59′	8	0.152	0.2778	.....	.....
6.8	51°41′	6°14′	4°52′	0.12821	0.8718	0.11177	0.9260	7.6253	0.9532	16°52′	8	0.147	0.2691	.....	.....
7.0	51°53′	6° 5′	4°45′	0.12500	0.8750	0.10938	0.9280	7.8280	0.9785	7°45′	8	0.143	0.2610	.....	.....
7.2	52° 5′	5°56′	4°38′	0.12195	0.8781	0.10708	0.9300	8.1331	0.9037	34°39′	9	0.139	0.2533	.....	.....
7.4	52°16′	5°48′	4°32′	0.11905	0.8809	0.10487	0.9319	8.3350	0.9261	26°34′	9	0.135	0.2461	.....	.....
7.6	52°27′	5°40′	4°26′	0.11628	0.8837	0.10276	0.9337	8.5375	0.9487	18°29′	9	0.132	0.2393	.....	.....
7.8	52°37′	5°32′	4°20′	0.11364	0.8864	0.10072	0.9353	8.7408	0.9712	10°23′	9	0.128	0.2329	.....	.....
8.0	52°47′	5°25′	4°15′	0.11111	0.8889	0.09877	0.9370	8.9436	0.9937	2°16′	9	0.125	0.2268	.....	.....
8.2	52°56′	5°18′	4° 9′	0.10870	0.8913	0.09688	0.9385	9.2491	0.9250	27° 2′	10	0.122	0.2210	.....	.....
8.4	53° 5′	5°11′	4° 4′	0.10638	0.8936	0.09507	0.9400	9.4530	0.9453	19°44′	10	0.119	0.2154	.....	.....
8.6	53°13′	5° 5′	3°50′	0.10417	0.8958	0.09332	0.9413	9.6544	0.9654	12°26′	10	0.116	0.2102	.....	.....
8.8	53°21′	4°59′	3°54′	0.10204	0.8980	0.09163	0.9427	9.8570	0.9857	5° 8′	10	0.114	0.2052	.....	.....
∞	60°	0°	0°	0	1	0	1	∞	0.	0°	∞	0	0	.....	.....

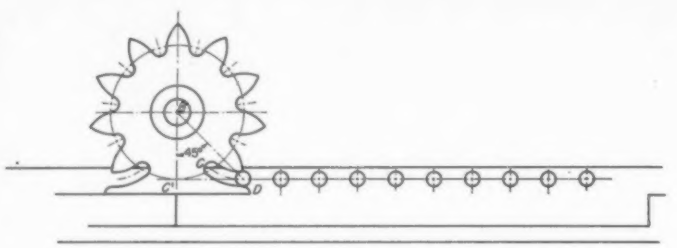
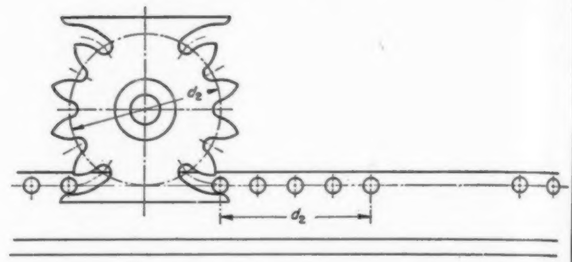


Fig. 22—Star wheel mechanism, $n=1$, $\mu=0$, $\epsilon=0$

Fig. 23—Star wheel mechanism, $n=2$, $\mu=0$, $\epsilon=0$



tion is simplified. In the many cases where a slight departure from the required value of ϵ is unobjectionable, it is advisable to determine a round, or at least a rational value for μ, that is very near to the required value.

EXAMPLE 5: Gears are to be designed for imparting one full revolution to the driven shaft during a third revolution of the driving shaft. The ratio ϵ is $\frac{1}{3}$, and n is 1. TABLE 5 indicates $\mu = 0.3027$. It will be found by means of a slide rule, or by means of a table of decimal equivalents of fractions, that the ratio $23/76 = 0.30263$ answers the requirements very closely.

The duration of motion, expressed as a fraction of one revolution of the driving gear, is

$$\nu = \frac{2\,\alpha_0 + \mu\left(\dfrac{2\,\pi}{n} - 2\,\beta_0\right)}{2\,\pi}$$

and since

$$\frac{2\,\alpha_0 + \mu\left(\dfrac{2\,\pi}{n} - 2\,\beta_0\right)}{\dfrac{2\,\pi}{n}} = \epsilon$$

it follows that

$$\nu = \frac{\epsilon}{n} \qquad (35)$$

In the case of $n = 1$,

$$\nu_1 = \epsilon_1 \qquad (36)$$

The extreme values of ν are

$$\nu_{max} = \frac{\epsilon_{max}}{n} \qquad (37)$$

$$\nu_{min} = \frac{\epsilon_{min}}{n} \qquad (38)$$

It has been found that the motion occupies the angle $2\alpha_0 + \mu[(2\pi/n) - 2\beta_0]$ of the driving gear. Therefore, the driven gear must be locked during the remaining part of one revolution, that is, during the angle

$$\gamma = 2\pi - \left[\, 2\,\alpha_0 + \mu\left(\frac{2\,\pi}{n} - 2\,\beta_0\right)\right]$$

$$= 2\left[\,\pi\left(1 - \frac{\mu}{n}\right) - \alpha_0 + \mu\,\beta_0\,\right] \qquad (39)$$

In the case of $n = 1$,

$$\gamma_1 = 2\,[\pi(1-\mu) - \alpha_0 + \mu\,\beta_0] \qquad (40)$$

The angles γ and γ_1 are also the angles included by the locking drum.

As with Geneva mechanisms, the radius of the locking drum can be chosen arbitrarily.

If μ has been determined, as in Example 5, and the center distance has been chosen, the design data— α_0, β_0, ϕ_0, r_1 r_2, s, and ρ_0—are either evaluated by means of the respective equations, or taken from TABLE 6 which contains these values in narrow steps of μ. Intermediate values can be obtained by linear interpolation. All these design data are independent of the number of stations n.

The ratio of gearing (ϵ), and the proportion of motion in one revolution of the driving shaft (ν), as well as the angle over which the locking drum extends (γ), depend on the number of stations, and may be evaluated from Equations 28, 35 and 39.

EXAMPLE 6: Design a star wheel mechanism which imparts, during a quarter revolution of the driving gear, a quarter revolution to the driven gear. Center distance $a = 8$ inches. A mechanism of that description is shown in *Fig.* 18. TABLE 5 indicates, for $\epsilon = 1$ and $n = 4$, the value $\mu = 0.7773$; a replacement by $7/9 = 0.7778$ is unobjectionable.

From Equation 20, $\sin \alpha_0/2 = (7/9)/2(16/9) = 7/32 = 0.21875$; $\alpha_0/2 = 12$ deg 38 min $= 12.633$ deg; $\alpha_0 = 25$ deg 16 min.

From Equation 22, $\phi_0 = 45$ deg $- \frac{3}{4}\alpha_0 = 45$ deg $-$ 18 deg 57 min $=$ 26 deg 3 min.

From Equation 23, $\beta_0 = 90$ deg $- (39/7)(12.633)$ deg $= 90$ deg $- 70.386$ deg $= 19.614$ deg $= 19$ deg 37 min.

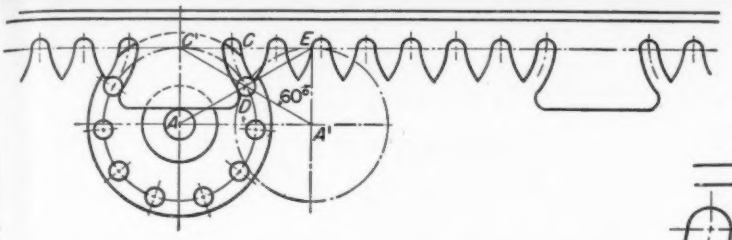


Fig. 24—Star wheel mechanism, $n = \infty$, $\mu = \infty$, $\epsilon = \epsilon_{max}$

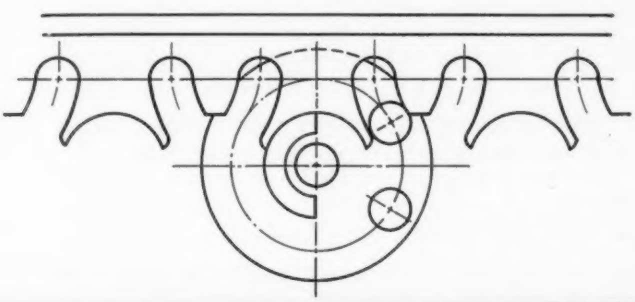


Fig. 25—Star wheel mechanism, $n = \infty$, $\mu = \infty$, $\epsilon = \epsilon_{min}$

From Equation 24, $r_1 = 8/[1 + (7/9)] = 72/16 = 4.5$ inches.

From Equation 25, $r_2 = 8(7/9)/[1 + (7/9)] = 56/16 = 3.5$ inches.

From Equation 26, $s = 8(7/9)/[1 + (7/9)]^2 = 504/256 = 1.969$ inches.

From Equation 27, $\rho_0 = 2(8)(7/16) \cos 51$ deg 19 min $= 4.375$ inches.

All these values can also be obtained from TABLE 6 by interpolation.

From Equation 28, $\epsilon = (7/9) + 4[4 + 3(7/9)] \times (12.633/180) - (4/2)(7/9) = 1.002$. The discrepancy of ϵ with the assumption of $\epsilon = 1$, is due to the modification of μ, and means that the motion occupies 90 deg 1 min instead of 90 deg.

From Equation 35, $\nu = \frac{1}{4}$, if the slight inaccuracy of ϵ is ignored.

From Equation 39,

$$\gamma = 2\left\{ 180\left[1 - \frac{7}{9(4)} \right] - 2(12.633) + \frac{7(19.614)}{9} \right\} \text{deg}$$

$$= 2(180 - 35 - 25.267 + 15.255) \text{ deg.} = 269 \text{ deg } 59 \text{ min}$$

which differs from the correct angle of 270 deg by the same slight amount as the angle of motion above. The angle can be made exactly 270 deg.

Relevant dimensions in the cases of $\mu = 0$ and $\mu = \infty$ are not fully determined in TABLE 6.

For $\mu = 0$, the center line of the slots is a part of an involute. The angle α_0 ($C'AD$ in *Fig.* 21*a*) is zero. The distance $C'D$ in *Fig.* 22 can easily be determined, nevertheless. Since the angle $C'BD = 45$ deg, $C'D$ is equal to the radius r_2 of the driven gear. The same result is also obtained by the graphical determination of $s = C'D$. Readers who find solution for the angle β_0 rather odd for $\mu = 0$ may be reminded that Equation 23 can be developed to the form, $\beta_0 = \frac{1}{2}\pi - \frac{1}{2}\alpha_0 - \frac{1}{2}\alpha_0/\sin \frac{1}{2}\alpha_0$, which takes on, for $\alpha_0 = 0$, the value $\frac{1}{2}\pi - 1$ radians, or 90 deg $- 180/\pi$ deg $=$ 90 deg $-$ 57 deg 18 min $=$ 32 deg 42 min.

For $\mu = 0$, the angle γ of the locking drum is zero. The locking shoe is flat, and its length can be chosen to suit requirements. It is, therefore, also possible to omit the means for locking altogether, and to let the accelerating roller commence acting at the moment when the retarding roller is leaving its slot. Such a case is shown in *Fig.* 23. Designing and machining a gear as shown in *Fig.* 23 ($\mu = 9$, $n = 2$) are simplified by the circumstance that the distance between the centers of the first and last roller of each group is equal to the pitch diameter of the driven gear.

It has already been found that μ can assume zero value only if n is 1 or 2 . *Figs.* 22 and 23 thus are the only possible embodiments, disregarding modifications in the duration of locking.

For $\mu = \infty$, the center line of the slots is a part of a cycloid. The angle β_0 ($C'BD$ in *Fig.* 21*a*) is zero, and the distance $C'C$ in *Fig.* 24 is $r_1 [\sqrt{3} - (\pi/3)] = 0.68485\, r_1$, wherein r_1 is the pitch radius of the driving gear. That relation can easily be vertified since the distance $C'E = \sqrt{3}\, r_1$, and the distance $CE = r_1\pi/3$.

A mechanism for the extreme case without standstills is shown in *Fig.* 24. *Fig.* 25 shows the other extreme where the period of uniform motion is reduced to a minimum, that is, to $\alpha_0 = 60$ deg. Motion and standstill occupy 180 deg each.

KINEMATICS: In *Fig.* 21*b* the star wheel mechanism is shown in a general position during the period of acceleration. CD is a part of the epicycloid which forms the center line of the slot. Similarly, as in *Fig.* 21*a*, A' is the center of the circle, rolling of which generates the epicycloid CD, in the position which corresponds to the generation of point D.

The angle $\alpha = C'AD$ is the angle through which the driving roller must rotate up to the central position. The triangles $C'AD$ and $EA'D$ are congruent, and as the arcs $CE = DE = r_1\alpha$, the angle $CBE = r_1\alpha/r_2 = \alpha/\mu$.

The angle $\beta = C'EC$ is the angle through which the driven gear must rotate up to the central position. The angle $C'BE = \beta + (\alpha/\mu)$, and the angle $C'BD$ is half of it, that is, $[\beta + (\alpha/\mu)]/2$.

Since $ID = r_1 \sin \alpha$, and $IB = r_2 + r_1 - r_1 \cos \alpha$,

$$\tan \frac{\beta + \dfrac{\alpha}{\mu}}{2} = \frac{r_1 \sin \alpha}{r_2 + r_1 - r_1 \cos \alpha} = \frac{\sin \alpha}{\mu + 1 - \cos \alpha}$$

Therefore, the relation between β and α is

$$\beta = 2 \tan^{-1} \frac{\sin \alpha}{\mu + 1 - \cos \alpha} - \frac{\alpha}{\mu} \quad \text{.............. (41)}$$

If the angular velocity of the driving gear is $\omega = 1$, the angular velocity of the driven gear is obtained by differentiating β with respect to α.

$$\frac{d\beta}{d\alpha} = 2 \frac{(\mu + 1) \cos \alpha - 1}{(\mu + 1)^2 + 1 - 2 (\mu + 1) \cos \alpha} - \frac{1}{\mu} \quad (42)$$

The maximum angular velocity is reached when $\alpha = 0$, as can easily be gathered from *Fig. 21b*.

$$\left(\frac{d\beta}{d\alpha}\right)_0 = \frac{1}{\mu} \quad \text{............................ (43)}$$

That result is not surprising because the gears behave in the central position like usual spur gears of the ratio $r_2/r_1 = \mu$.

Differentiating Equation 42 with respect to α leads to the angular acceleration of the driven gear.

$$\frac{d^2\beta}{d\alpha^2} = -2 \mu (\mu + 1) (\mu + 2) \times \frac{\sin \alpha}{[(\mu + 1)^2 + 1 - 2 (\mu + 1) \cos \alpha]^2} \quad \text{............ (44)}$$

As with Geneva mechanisms, the minus sign indicates here also that the driven gear is accelerated before the central position.

The angular acceleration at the start of motion is obtained by substituting $(-\alpha_0)$ for α in Equation 44. Since $\sin \alpha_0/2 = \mu/2(1 + \mu)$, it follows that

$$\cos \frac{\alpha_0}{2} = \sqrt{1 - \sin^2 \frac{\alpha_0}{2}} = \frac{\sqrt{(\mu + 2)(3\mu + 2)}}{2 (\mu + 1)}$$

$$\sin (-\alpha_0) = -2 \sin \frac{\alpha_0}{2} \cos \frac{\alpha_0}{2} = -\frac{\mu \sqrt{(\mu + 2)(3\mu + 2)}}{(\mu + 1)^2}$$

$$\cos (-\alpha_0) = 1 - 2 \sin^2 \frac{\alpha_0}{2} = \frac{\mu^2 + 4\mu + 2}{2 (\mu + 1)^2}$$

Inserting the values of $\sin(-\alpha_0)$ and $\cos(-\alpha_0)$ in Equation 44 leads to

$$\left(\frac{d^2\beta}{d\alpha^2}\right)_{-\alpha 0} = \frac{\mu + 1}{\mu^2} \sqrt{\frac{3 \mu + 2}{\mu + 2}} \quad \text{.................. (45)}$$

The third differential coefficient is

$$\frac{d^3\beta}{d\alpha^3} = -2 \mu (\mu + 1) (\mu + 2) \times \frac{2(\mu+1) \cos^2 \alpha + (2 + 2\mu + \mu^2) \cos \alpha - 4 (\mu + 1)}{[(\mu + 1)^2 + 1 - 2 (\mu + 1) \cos \alpha]^3} \quad \text{.. (46)}$$

The maximum of $d^2\beta/d\alpha^2$ pertains to the position where $d^3\beta/d\alpha^3 = 0$, or where

$$2(\mu + 1) \cos^2 \alpha + (2 + 2 \mu + \mu^2) \cos \alpha - 4 (\mu + 1) = 0$$

The solution is

$$\cos \alpha = -\frac{\mu^2 + 2 (\mu + 1)}{4 (\mu + 1)} \pm \sqrt{\left[\frac{\mu^2 + 2 (\mu + 1)}{4 (\mu + 1)}\right]^2 + 2}$$

The value under the square root is larger than 1, and only the plus sign is to be used so that

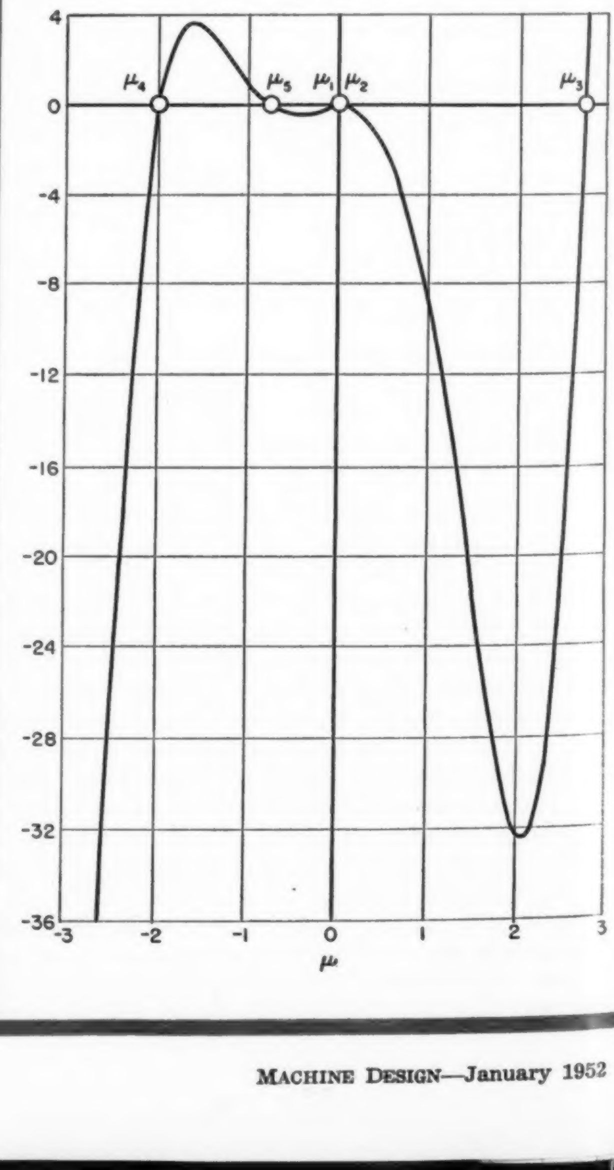
$$\alpha_{max} = \cos^{-1} \left\{ -\frac{\mu^2 + 2 (\mu + 1)}{4 (\mu + 1)} + \sqrt{\left[\frac{\mu^2 + 2 (\mu + 1)}{4 (\mu + 1)}\right]^2 + 2} \right\} \quad \text{............. (47)}$$

The maximum of the angular acceleration is obtained by substituting $\alpha = \alpha_{max}$ in Equation 44.

$$\left(\frac{d^2\beta}{d\alpha^2}\right)_{max} = -2 \mu (\mu + 1) (\mu + 2) \times \frac{\sin \alpha_{max}}{[(\mu + 1)^2 + 1 - 2 (\mu + 1) \cos \alpha_{max}]^2} \quad \text{....... (48)}$$

The angle α in all previous equations is not limited, from the mathematical point of view, to the period of motion. The existence of a maximum of $d^2\beta/d\alpha^2$ does not, therefore, necessarily mean that the maximum pertains to an actual position during the period of acceleration. The maximum is only of interest if

Fig. 26—Graph of $y = \mu^2 (\mu^3 - 6\mu - 4)$



$\alpha_{max} < \alpha_0$, or if $\cos \alpha_{max} > \cos \alpha_0$. From Equation 47 and the relation $\cos \alpha_0 = (\mu^2 + 4\mu + 2)/(\mu + 1)^2$, it follows by inspection that

$$-\frac{\mu^2 + 2(\mu + 1)}{4(\mu + 1)} + \sqrt{\left[\frac{\mu^2 + 2(\mu + 1)}{4(\mu + 1)}\right]^2 + 2} > \frac{\mu^2 + 4\mu + 2}{2(\mu + 1)^2}$$

Isolating the square root on one side and squaring both sides of this inequality give the more convenient form, $\mu^2(\mu^3 - 6\mu - 4) < 0$.

The curve $y = \mu^2(\mu^3 - 6\mu - 4)$ is plotted in *Fig.* 26. The factor μ^2 indicates that two of the five solutions are $\mu_1 = \mu_2 = 0$.

The origin is a double point of the curve, and the μ axis is the tangent at that point. The other three solutions are the roots of the cubic equation $\mu^3 - 6\mu - 4 = 0$. They are

$$\mu_3 = 2\sqrt{2}\cos 15 \text{ deg} = 1 + \sqrt{3}$$
$$= 2.732$$

$$\mu_4 = -2\sqrt{2}\cos 45 \text{ deg}$$
$$= -2$$
$$\mu_5 = -2\sqrt{2}\cos 75 \text{ deg} = 1 - \sqrt{3}$$
$$= -0.732$$

If only the values of $\mu \geqq 0$ are contemplated, the condition $\alpha_{max} < \alpha_0$ is complied with if $\mu^2(\mu^3 - 6\mu - 4)$ is negative, that is, for $0 < \mu < 2.732$. For $\mu < 2.732$, the acceleration reaches its maximum within the actual period of acceleration. For $\mu > 2.732$, the maximum acceleration pertains to an angle which lies befor the commencement of motion, and the largest value is that at the commencement of motion.

The value of $d^3\beta/d\alpha^3$ for $\alpha = 0$, is $-(\mu+1)(\mu+2) \div \mu^3$. It is a measure for the rate at which the angular acceleration diminishes. However, it is of lesser interest than the analogous value of Geneva mechanisms, because the insertion of uniform motion between the periods of acceleration and retardation does not cause the driving roller to change contact from one flank of the slot to the other.

In *Fig.* 27 are shown diagrams of β, $d\beta/d\alpha$ and $d^2\beta/d\alpha^2$ for the mechanism shown in *Fig.* 18, which

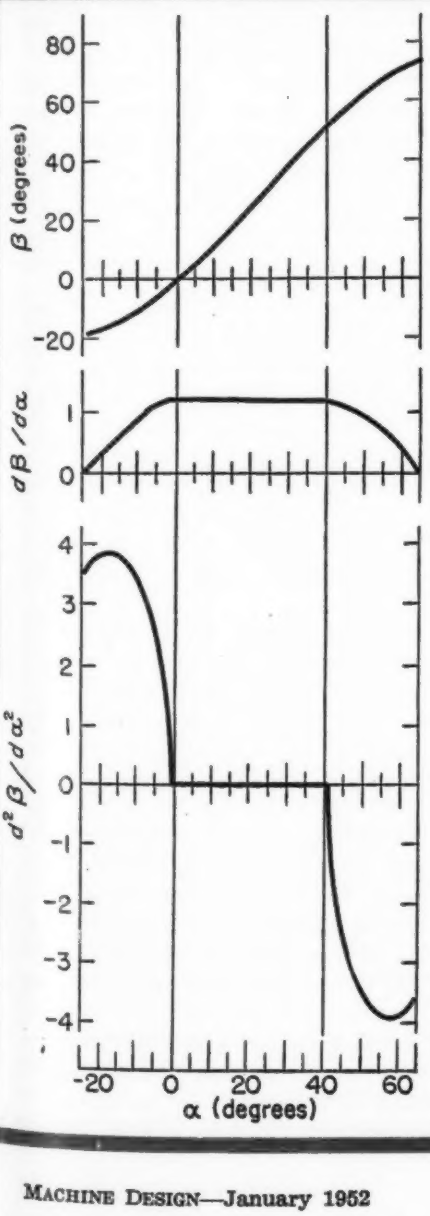
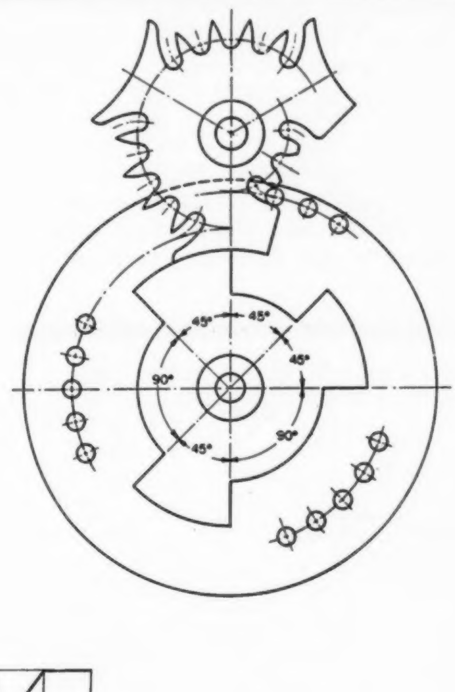


Fig. 27—Left—Diagram of motion for the mechanism shown in *Fig.* 18

Fig. 28—Right—Modified star wheel mechanism with varying periods of motion



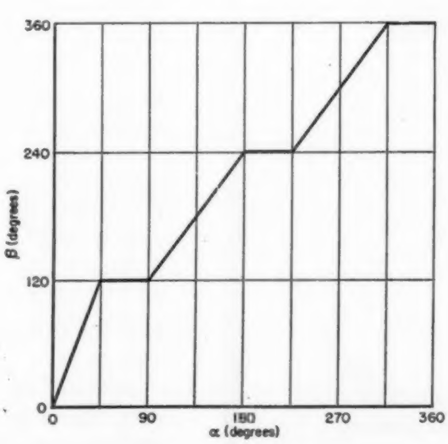


Fig. 29—Left—Displacement diagram for the mechanism of *Fig.* 28

has been considered in Example 6. The angular acceleration starts at $\alpha_0 = -25$ deg 16 min with a definite amount, rises to a maximum at $\alpha_{max} = -19$ deg 1 min, and decreases until it reaches zero at $\alpha = 0$. The motion is uniform during the subsequent period of 39 deg 28 min. The angular retardation during the following period lasting 25 deg 16 min is analogous to the angular acceleration in the first period.

Contained in TABLE 6 are the values $(d\beta/d\alpha)_0$, $(d^2\beta/d\alpha^2)_{-\alpha_0}$, α_{max}, and $(d^2\beta/d\alpha^2)_{max}$, the two latter only for $\mu < 2.732$.

EXAMPLE 7: Investigate the kinematic properties of the mechanism shown in *Fig.* 18. The geometrical relations of that mechanism have been considered in Example 6, where μ has been assumed 7/9. $N = 50$ rpm, that is, the same speed as considered in Example 1 for an equivalent Geneva mechanism.

The maximum angular velocity, for $\omega = 1$, is according to Equation 43, $1/\mu = 9/7 = 1.286$ rad per sec. For $N = 50$ rpm, the angular velocity is 5.2360 (1.286) = 6.732 rad per sec.

The angular acceleration at the start, for $\omega = 1$, is according to Equation 45, 3.670 rad per sec^2. For $N = 50$ rpm, it is 27.416(3.670) = 100.617 rad per sec^2.

The maximum angular acceleration pertains to $\alpha_{max} = -19$ deg 1 min, obtained from Equation 47, and is 3.921 rad per sec^2, obtained from Equation 48. For $N = 50$ rpm, it is 27.416(3.921) = 107.497 rad per sec^2.

A comparison between the results of Examples 1 and 7 reveals that a star wheel mechanism with $n = 4$ and $\epsilon = 1$, has more favorable kinematic conditions than an equivalent Geneva mechanism. The same will be found to a still higher degree for $n = 3$, and to lesser degree for $n = 5$.

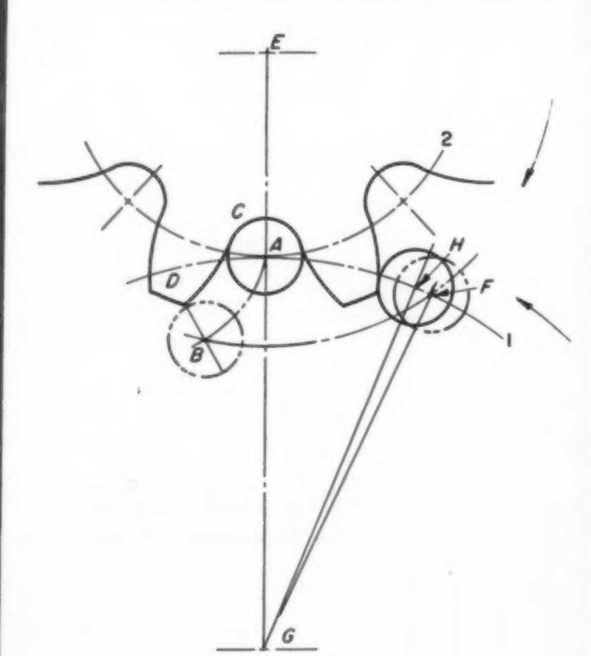


Fig. 30—Graphical determination of duration of engagement in pin gearing

For $n \geqq 6$ however, the maximum acceleration of Geneva mechanisms is lower than in equivalent star wheel mechanisms. That is due to the circumstance that Geneva mechanisms have longer periods of acceleration, so that the average acceleration is smaller than in the case of star wheel mechanisms. Only the unfavorable course of motion of Geneva mechanisms with a small number of stations is responsible for the high peaks of the acceleration.

MODIFICATIONS: Similar to those described for external Geneva mechanism in the first article of this series, modifications can also be applied to external star wheel mechanisms.

The relation between the number m of equally distributed groups of driving rollers and the ratio ν is, as with modified Geneva mechanisms, $m\nu \leqq 1$, or $m \leqq 1/\nu$. From the relation $\nu = \epsilon/n$ (Equation 35), it follows that $m \leqq n/\epsilon$. The maximum of m thus pertains to the minimum of ϵ.

$$m_{max} = \frac{n}{\epsilon_{min}} \quad \text{(49)}$$

The nearest smaller whole number to the result of Equation 49 is to be found in TABLE 4 as m_{max}. These values are higher, or at least as high as the analogous values for Geneva mechanisms. Two groups of driving rollers can be arranged for any number of stations. If $\epsilon_{min} = 3n\alpha_{0min}/2\pi$ (Equation 32) is used together with Equation 49, it follows that

$$m_{max} = \frac{2\pi}{3\,\alpha_{0-min}} \quad \text{(50)}$$

Angle α_0 is for finite numbers of stations less than 60 deg $= \pi/3$, so that $m_{max} = (2\pi/3)(3/\pi) = 2$.

The driving rollers can be arranged in equally or unequally spaced groups. *Fig.* 17, for example, shows in dotted lines two additional groups of driving rollers which would let the driven gear make half a revolution during every third revolution of the driving gear. If only one of the additional groups is provided, the unequally spaced groups of driving rollers get into action alternately after $\frac{1}{3}$ and $\frac{2}{3}$ revolutions of the driving gear.

In *Fig.* 28 is shown a mechanism for imparting intermittent motion with varying duration of motion, and *Fig.* 29 shows the corresponding displacement diagram schematically. The driven gear makes three partial movements, each of 120 deg, divided by three standstills lasting 45 deg of the driving gear. One of the partial movements lasts 45 deg, and the two others 90 deg of the driving gear. For the partial movement which lasts 45 deg, $\epsilon = 45/120 = 0.375$, and TABLE 5 indicates, for $n = 3$, the value $\mu = 0.2869$. For the partial movements lasting 90 deg, $\epsilon = 90/120 = 0.75$, and $\mu = 0.6047$. The pitch radii and all other relevant dimensions vary with varying values of μ.

Mechanisms can be laid out in a similar way not only for varying duration of motion and standstill, but also varying partial movements.

DESIGN AND PRODUCTION: The first section of this

article refers to the elements for imparting the nonuniform motion. The angles β_0 and ϕ_0, and the distance ρ_0 determine the slot of the driven gear, and the tangents at both ends of the slot enable the center line of the slots to be drawn almost without constructing the epicycloid.

The angle between accelerating and retarding rollers (i.e., the first and last roller of each group) is $\mu[(2\pi/n) - 2\beta_0]$. The angle over which the locking drum extends is the angle γ which is defined by Equation 39.

The elements for imparting uniform motion must be laid out according to the rules of common gearing. Pin gearing is chosen in most of the illustrations mainly because it renders them self-explanatory. In actual practice pin gearing has also the advantage (probably the only one) of a certain simplicity. The rollers for accelerating, imparting uniform motion, and retarding, are uniform and arranged on a common disk, and the slots for nonuniform motion and the teeth for uniform motion are in the same plane. Some indications for pin gearing will be given, as far as they are required for the purpose in question.

The relative path along which the center of a driving roller travels in the system of the driven gear is a part of an epicycloid, congruent with that used for the slots for nonuniform motion.

The distance between the accelerating roller and its neighbor can be determined by the following consideration. The acceleration occupies the angle α_0 of the driving gear, and the accelerating roller leaves its slot after a rotation through a further angle α_0. The driving gear must therefore rotate, between the start of motion and the position where the uniform motion parts come into action, through an angle which lies between α_0 and $2\alpha_0$. The pitch of the intermediate rollers can be chosen arbitrarily within certain limits. There is a lower limit, since with decreasing pitch the teeth become weaker, and an upper limit is drawn by the requirement that at any time at least two of these rollers must be in action.

The angle during which a roller is active can be found graphically as shown in *Fig.* 30. *AB* is a part of the epicycloid which is generated by the rolling of the pitch circle of the driving gear along the pitch circle of the driven gear. The flank *CD* is an equidistant curve at the distance of the radius of the roller. The flank does not exactly represent near the root the shape intended by the epicycloidal center line, because the radius of curvature there is smaller than the radius of the roller.

With the indicated direction of rotation, the roller is in the central position *A* at the end of engagement with the flank *CD*.

The height of the tooth determines the extreme point *D* of the flank, and that point pertains to the point *B* on the epicycloid A circle through the point *B* round the center *E* of the driven gear determines at its intersection *F* with the pitch circle 1 the commencement of engagement.

If it is required that the second roller starts its action a certain angle (for example, α_0) after the accelerating roller, the angle *AGF* is made that angle, and the height of teeth determined by reversing the construction. If the height of the teeth serving for the transmission of uniform motion is chosen equal to the height of the first and last tooth, the pitch of the intermediate teeth must be smaller than the arc *AF*, in order to have two rollers simultaneously in action. The pitch *AH* in *Fig.* 30 is shown smaller than the arc *AF*. If the pitch of all rollers is chosen uniform, the height of the intermediate teeth must exceed that of the first and last tooth.

EXAMPLE 8: Design the uniform motion parts of a star wheel mechanism for $n = 1$ and $\mu = 0.375$ (*Fig.* 16) . The center distance is $5\frac{1}{2}$ inches, and the diameter of the rollers may be chosen $\frac{1}{2}$ inch.

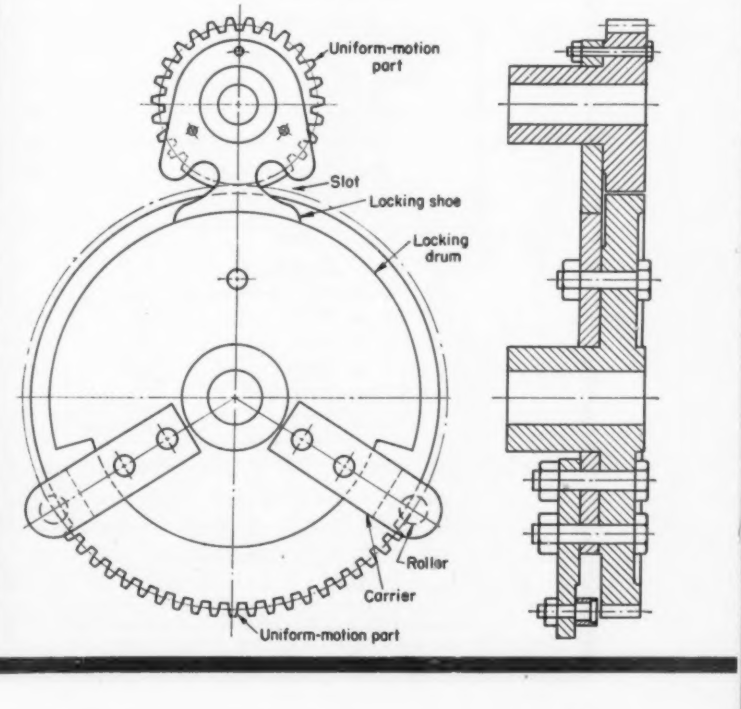
For $\mu = 0.375$, the angle $\alpha_0 = 15$ deg $40\frac{1}{2}$ min, the angle $\beta_0 = 24$ deg 41 min, the pitch radii are $r_1 = 4$ inches and $r_2 = 1\frac{1}{2}$ inches.

The uniform motion occupies 360 deg $- 2\beta_0 = 310$ deg 38 min of the driven gear, or 0.375 (310 deg 38 min) $=$ 116 deg 29 min of the driving gear. If the height of the teeth is chosen $\frac{5}{16}$ inch, it will be found by proceeding as demonstrated in *Fig.* 30, that a roller remains in action during $16\frac{1}{2}$ deg. Since that angle lies between α_0 and $2\alpha_0$, the height of teeth is suitable.

If, as in *Fig.* 16, eight equally spaced pins or rollers are chosen, the angle between two consecutive pins is (310 deg 38 min)/7 = 44 deg $22\frac{1}{2}$ min. The central teeth must be higher than the two adjacent to the accelerating and retarding slots.

It has already been mentioned that the uniform motion can be obtained by common involute gears, and *Fig.* 31 shows the mechanism of *Fig.* 16 carried out in that way. The locking drum is secured to the uniform motion part of the driving gear, and the part with locking shoe and slots is secured to the uniform motion part of the driven gear. The driving

Fig. 31—Combination of pin and involute gearing for a mechanism equivalent to that shown in *Fig.* 16



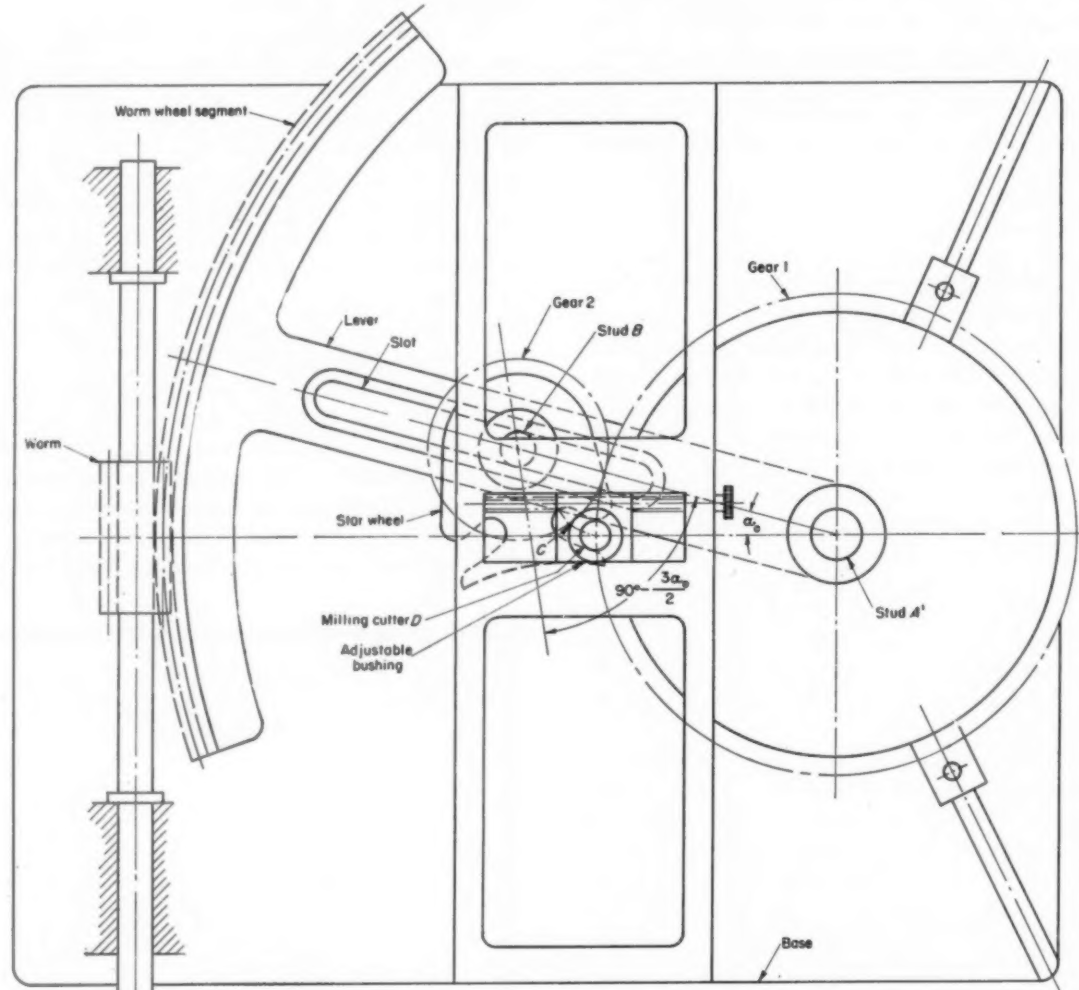
unit incorporates, moreover, the carriers for the rollers.

The angle δ of approach for involute gears (i.e., the angle between the position of the driving gear where a tooth first meets a mating tooth, and the position where the contact takes place in the central position) can be computed from the overall radius e_2 of the driven gear, the pitch radius r_1 of the driving gear, the pressure angle ψ, and the ratio $\mu = r_2/r_1$ of the pitch diameters. It may be stated without proof that the angle of approach δ in radians is

$$\delta = \sqrt{\left(\frac{e_2}{r_1 \cos \psi}\right)^2 - \mu^2} - \mu \tan \psi \quad \ldots\ldots\ldots\ldots (51)$$

The angle between the center of the accelerating roller and the tip of the first involute tooth must be at least the angle δ defined by Equation 51. The number of teeth in the involute gear segment can be calculated if the duration of uniform motion and the

Fig. 32—Device for milling epicycloidal slots



angle δ are known. It can also be found graphically.

Example 9: Design the uniform motion parts of a star wheel mechanism for $n = 1$ and $\mu = 0.375$, as shown in *Fig.* 31. The center distance is $5\frac{1}{2}$ inches.

An equivalent mechanism has been considered in Example 8, and the pitch radii have been stated there as $r_1 = 4$ inches, and $r_2 = 1\frac{1}{2}$ inches. If the diametral pitch of the involute gears is chosen $P = 9$, the complete gears would have 72 and 27 teeth, respectively. If the height of teeth above pitch line is the usual amount of $1/P$, the overall radius of the driven gear is $e_2 = 29/18 = 1.611$ inches; $r_1 = 4$ inches; ψ may be assumed $14\frac{1}{2}$ deg, according to an available milling cutter or hob. $\mu = 0.375$. Equation 51 leads to

$$\delta = \sqrt{\left(\frac{1.611}{4\cos 14\frac{1}{2}}\right)^2 - 0.375^2} - 0.375\tan 14\tfrac{1}{2}$$

$$= 0.08319 \text{ rad} = 4 \text{ deg } 46 \text{ min}$$

The uniform motion has been found in Example 8 to occupy 116 deg 29 min of the driving gear, so that the angle during which the involute gears are effective should be less than 116 deg 29 min − 2 (4 deg 46 min) = 106 deg 57 min. The pitch angle is 360/72 = 5 deg, so that 106 deg 57 min corresponds to 21.39 pitches. The arc covered by 22 teeth, measured along the pitch circle between the respective external flanks, is $21\frac{1}{2}$ pitches, and the distance of the external tips of the two extreme teeth corresponds to a somewhat smaller angle. Twenty-two teeth could be possible, but to be on the safe side, 21 teeth have been chosen in *Fig.* 31. Since there is an odd number of teeth, the driving gear has a tooth, and the driven gear has a gap in the central position. Only 21 gaps need be machined in the driven gear, corresponding to the same number of teeth in the driving gear. However, there is no objection against cutting all teeth of the driven gear if the teeth are cut by hobbing.

The previous example makes it obvious that the pitch diameter of driving and driven gears, which are common to uniform and nonuniform motion, must be chosen to suit the requirements of involute gearing. Therefore, the design is best started from a suitable value of μ.

If involute gears are used for imparting the uniform motion, the only parts which involve some difficulty in machining are the accelerating and retarding slots. The slots can be marked off, and drilled and filed to shape. Where star wheels are required in a considerable quantity, an attachment as shown in *Fig.* 32 simplifies the work. The base is equipped with stud A', on which gear 1 is rigidly mounted. Gear 1 must have the same pitch diameter as the driving gear, and the driving gear which imparts the uniform motion can be used advantageously. A lever, pivoting round stud A' has a slot, and terminates in a segment of a worm wheel which meshes with a worm operated by a crank. Stud B is adjusted in the slot of the lever so that the distance between studs A' and B is equal to the center distance of the star wheel mechanism to be machined. Gear 2 of the pitch diameter of the driven gear pivots round stud B, and meshes with gear 1. Here again, the uniform motion part of the driven gear can be used as gear 2.

A milling cutter of the diameter of the driving roller is guided in an adjustable bushing, and the milling attachment is mounted in a way that the distance between stud A' and the milling cutter is equal to the pitch radius of the driving gear.

The angle $DA'B$ is made α_0, and the blank in which the slot is to be machined is connected with gear 2 in a way that the center line includes with the center line of the lever the angle 90 deg − $(3\alpha_0/2)$.

It will be found that the points A', B, D, and E are identical with the analogous points in *Fig.* 21*a*, and on turning the crank the milling cutter produces the required slot along the center line DC. Pointers (not shown) may simplify the adjustments of the two angles required.

Manufacture of gears like those shown in *Figs.* 20 and 25 is simplified by the absence of special elements for importing uniform motion. Accelerating and retarding rollers include, in these cases, the angle α_0. The retarding roller enters its slot when the accelerating roller reaches the central position, and both rollers impart uniform motion during an angle α_0 of the driving gear. Elements for imparting uniform motion can also be dispensed with if the angle between the accelerating and retarding roller is increased up to $2\alpha_0$, in which case the retarding roller enters in the moment when the accelerating roller leaves its slot. As the duration of uniform motion is $\mu[(2\pi/n) - 2\beta_0]$, special elements for imparting uniform motion are unnecessary if $\mu[(2\pi/n) - 2\beta_0] < 2\alpha_0$. That condition can be transformed, by use of Equation 23, into $\alpha_0 < (\pi/3)(n - 2/n)$, and further, by use of Equation 20, into

$$\mu < \frac{2\sin\dfrac{\pi}{6}\,\dfrac{n-2}{n}}{1 - 2\sin\dfrac{\pi}{6}\,\dfrac{n-2}{n}} \quad \text{(53)}$$

For $n = 1$, we obtain $\mu < -0.5$; for $n = 2$, $\mu < 0$. As μ must be positive, for external star wheel mechanisms, uniform motion parts must be provided in any case of $n = 1$ or 2.

For $n \geqq 3$, Equation 52 leads to positive values of μ. Elements for imparting uniform motion can be dispended with between μ_{min} and the values obtained by Equation 52. These ranges are

n	μ Range
3	0.0241 to 0.5321
4	0.3943 to 1.0731
5	0.7696 to 1.6180
6	1.1483 to 2.1650
7	1.5280 to 2.7133
8	1.9092 to 3.2619
9	2.2911 to 3.8114
10	2.6731 to 4.3612

These ranges comprise only a small part of the full ranges of μ. Since the frequent cases of $n = 1$ and 2 are excluded altogether, the feature of accelerating and retarding rollers serving also for the uniform motion can be utilized in a restricted number of cases only.

Internal star wheel mechanisms will be discussed in the next part of this series.

Rubber-Metal Parts

How to design for satisfactory service and lowest production costs

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RUBBER can be bonded readily to metals during cure. Bonding is practiced commercially in the manufacture of engine and motor mountings which are used for vibration insulation and in many other mechanical structures where positive adhesion of rubber to metal is desirable, *Fig.* 1.

In the usual method of bonding rubber to metal for small molded parts the metal parts are first thoroughly cleaned and then electroplated with a brass alloy. This is followed by application of a piece of unvulcanized rubber compound of suitable size. The assembly is put into a heated mold and squeezed into shape. While under molding pressure, which may be several thousand pounds per square inch, the rubber in the assembly is vulcanized at a temperature which ordinarily does not exceed 325 F. The bond strength between metal and rubber develops during vulcanization.

Since the rubber member is vulcanized in a mold at high pressures and since it is necessary to allow for proper flow of the stock during molding in order to obtain a good bond, the most advisable and usually the most economical design is one in which the metal parts are plain with no projections on the outer surfaces of the member or extending into the rubber body. In some cases it is not possible to do away with projections into the rubber body but for various reasons they are not generally recommended. If it is necessary for mechanical reasons to have any projections on the outer surfaces of the member it is usually found more economical to make plain members with provision for the attachment of the projections after vulcanization. Such attachment should be effected by bolts, rivets or means other than welding, which is undesirable because the heat may injure the rubber or the bond.

Bond Stresses: Since rubber bonded to metal is put to such a diversity of uses and since the metal surfaces to which the rubber is bonded are often irregular, it is not possible to specify a set of maximum advisable bond stresses which apply to all practical cases. The bond stresses are fixed by the working stresses which are advisable or necessary for the part. And the working stresses themselves must be chosen by compromise so that they are low enough to insure good static and dynamic fatigue for the unit and high enough to insure a practical cost level.

Unless specifically stated otherwise, a tension or shear bond strength of 200 psi for 30 to 40 durometer stocks or 250 psi for stocks of higher durometers is generally expected for high-grade rubber compounds.

Ordinarily, the compromise choice of working stress will vary with the application. However, as a rough guide maximum bond stresses for various types of deformation are: (1) Compression—relatively high but shape dependent; (2) shear plus compression—70 psi; (3) shear plus tension—relatively low but shape dependent.

Fig. 1—Typical rubber-metal bonded parts designed to reduce noise and vibration

Compression stresses being shape dependent—that is, determined by the ratio of length or width to thickness—are best exemplified by a nomograph such as that shown in *Fig.* 2 which also permits determination of safe deflections.

In connection with the preceding discussion of maximum stresses there are two points which cannot be emphasized too strongly: (1) The interpretation of "maximum" as used here and (2) the importance of distinguishing between "maximum bond stresses," as tabulated, and "maximum advisable bond stresses." Many rubber units are made to be used in vibration. For steady-state vibration in a particular application the maximum bond stress occurs when the rubber has its maximum deflection. There is always the possibility that transient phenomena may occur and for a short time lift the bond stresses to values much higher than those for the steady-state. In designing a mounting, care should be taken that the maximum bond stresses, including any transient phenomena, do not exceed the allowable values. If the part is to last only a few months it is possible that even higher stresses than those indicated can be used safely. The most economical choice of working stress is that which gives the optimum balance between adequacy in service and economy in production. The advisable values of working stresses for any particular case depend on the service conditions, the shape of the unit, the contour of the rubber-to-metal interface, the stock used and the service life which is required.

A mounting should be so designed that under load there are no regions of high localized stress, particularly in the neighborhood of rubber-to-metal junction. It is not sufficient that the mounting be designed so that most of the bond area is stressed within an advisable working limit. If any portion of the bond area is seriously overstressed, the result will be essentially the same as if the whole bond area were under such stress.

Thus it is advisable to have no pronounced irregularities on the metal surfaces to which the rubber is bonded, such as bolt heads, bolt holes, indentations, projections, etc. A smooth metal surface is best for bonding rubber. Any corrugations, pitting, depressions or the like in the metal surfaces introduce high localized bond stresses and are not considered advisable. Where projections from the bonding surface into the rubber are unavoidable or expedient, it is advisable that the design of these projections be made so as to avoid stress concentrations. For example, peened-in bolts should have as thin a head as practicable and the edge of this head should be rounded.

In general, bending of the metal parts of a mounting under load reduces the average bond stress but is likely to introduce high localized bond stresses. Accordingly, care must be taken in designing the metal parts of the mounting so that the fundamental design assumptions as to maximum bond stress are not impaired by distortion of the metal parts.

In case the rubber itself is to be stressed higher than a limiting advisable bond stress, the mounting should be designed so that the plain smooth surfaces to which the rubber is bonded are sufficiently greater than the effective working cross-section of the rubber to reduce the bond stresses to advisable values.

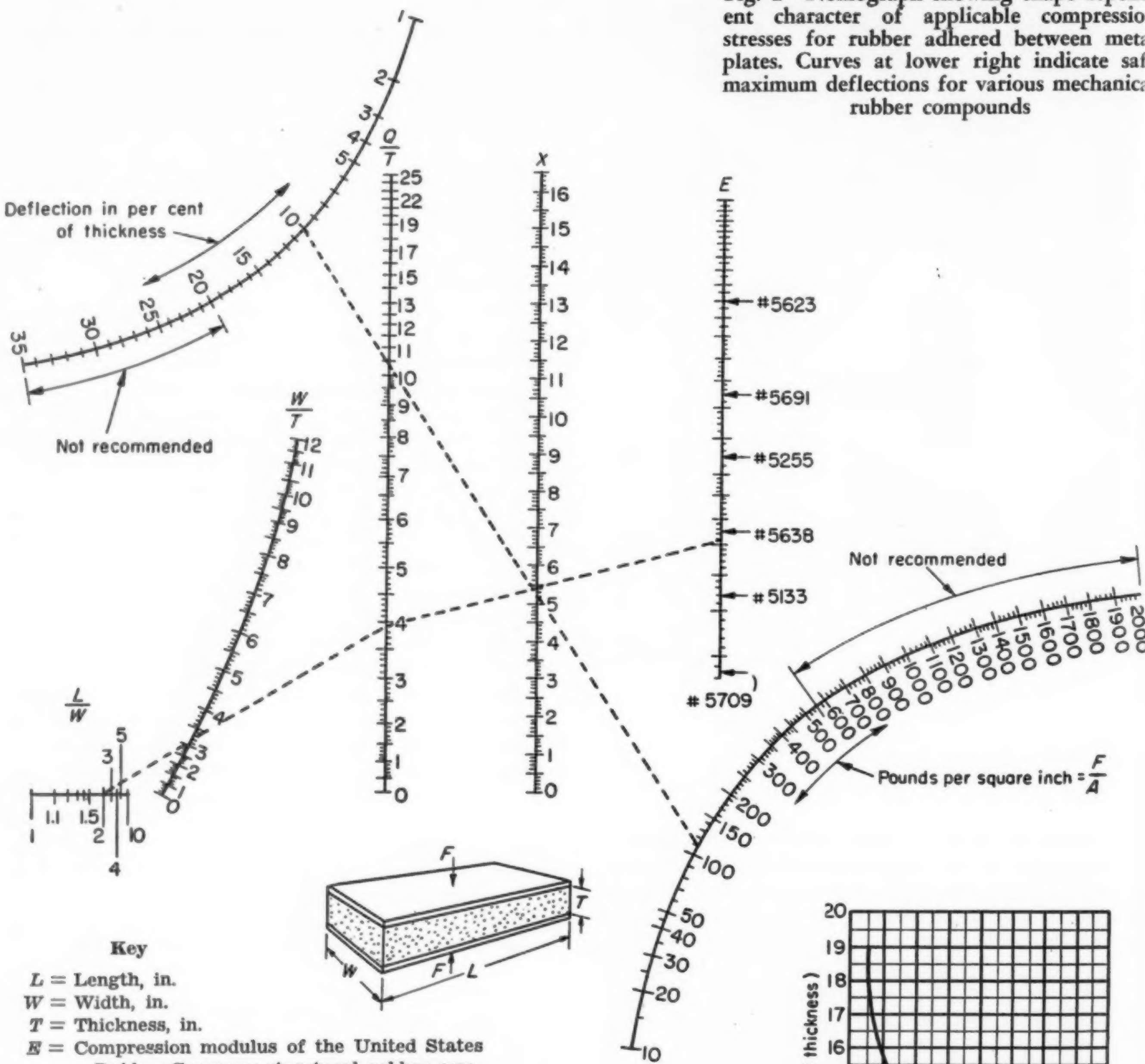
Low Carbon Steel Best

Type of Metals: The plating process can be applied to nearly all types of steel, nickel, zinc, copper, brass, Monel, and cast iron. In general, SAE 1010 hot-rolled, full-pickled steel is the most suitable metal for bonding; moreover, the bonding of rubber to this metal can be done at lower prices than bonding to other metals such as aluminum which require special processing. If a metal other than SAE 1010 hot-rolled steel is under consideration, it is advisable to submit a sample of the metal (with the metallurgical analysis and treatment) for adhesion tests to rubber.

Steel from which metal parts are made comes in several classifications. Strip, bar, sheet and plate are the ones commonly used for SAE 1010 hot-rolled steel. For the sake of economy the parts should be designed so that they can be made from steel in one of the standard classifications and sizes furnished by the steel industry.

Specifications: In order to obtain the economic advantages, quality and uniformity of the brass-plate

Fig. 2—Nomograph showing shape-dependent character of applicable compression stresses for rubber adhered between metal plates. Curves at lower right indicate safe maximum deflections for various mechanical rubber compounds



Key

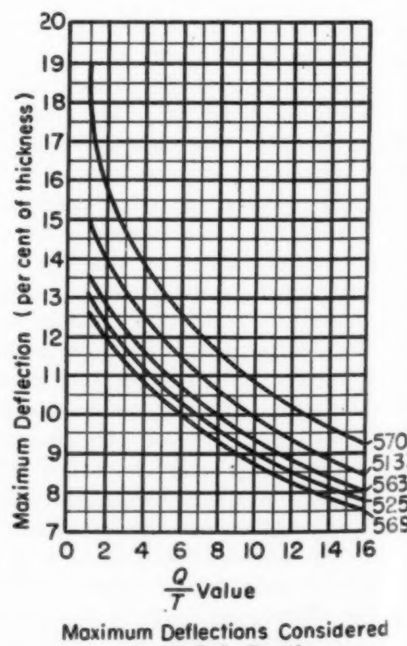
L = Length, in.
W = Width, in.
T = Thickness, in.
E = Compression modulus of the United States Rubber Company structural rubber compound indicated.
X = A reference line arbitrarily scaled for convenience.
F = Applied load, lb.; $\frac{F}{A}$ = unit load, psi.
$\frac{Q}{T}$ = Length of side of equivalent square divided by T.

Extension of straight line through ratios $\frac{L}{W}$ and $\frac{W}{T}$ determines a point on $\frac{Q}{T}$ scale.

Connect $\frac{Q}{T}$ and E by a straight line. Note point on scale X.

A straight line through this point on X connects respective values of unit load and per cent deflection.

For a square sandwich $L = W = Q$ and $\frac{Q}{T} = \frac{W}{T}$.



Maximum Deflections Considered to be Safe Practice

bonding process, the following specifications must be met:

1. The metal must be free from acid black
2. The metal must be free from rust, scale, slag inclusions and other foreign substances
3. If grease is used in making stampings, the grease should be saponifiable
4. All tapping should be done with soluble cutting oils
5. If possible no heavy oil should come in contact with the metals. In case this cannot be avoided, the oil must be removed to such an extent that specification 6 is satisfied
6. All metal must be of such a grade and so prepared that it can be cleaned easily in automatic pickling and cleaning units
7. The metal must be prime material. No mill rejects or seconds can be used.

Tolerances: Tolerances to which metal parts should be fabricated are:

1. Tolerances on outside dimensions of metal parts should never be plus. The usual tolerance in sizes up to 12 inches is plus 0.000, minus 0.015-inch. For pieces larger than 12 inches the allowable minus tolerance may be as high as 0.030-inch. (If components with too much tolerance are used with no means of definitely locating their position in the mold cavity they may "float" into undesirable positions)
2. Holes in metal parts through which molding cores or pins in the mold must pass either to prevent metal parts from floating during cure or to provide voids in the rubber should be $\frac{1}{32}$-inch larger in diameter than the nominal size desired and should be furnished to a tolerance of minus 0.000, plus 0.032-inch. Clearance hole tolerances of this type should never be minus. In cases where the customer's subsequent processing and assembling necessitates precision dimensions on metal parts, such as locating or bolting holes, these can be processed by allowing suitable mold clearances.

It is to be noted that the usual tolerances for commercial metal parts such as drawn stampings, sheared or blanked metal parts, and forgings or castings are greater than those allowed in the above specifications.

Channel-shape sections are frequently used in rubber and metal adhesion parts. Unless the metals are properly fabricated they give considerable trouble in production. *Fig.* 3 indicates important points to watch. Suggestions relative to a hole through a rubber-metal part are illustrated by *Fig.* 4.

In cases where the molding pressure is to be maintained by means of contact of the metal parts with the machined surfaces of the mold it is absolutely necessary that the contact surfaces of the metal parts be machined. If a casting or forging is to be used as a shell surrounding a rubber and metal part it is desirable that it be completely machined on the outside surface. If machined bosses are used as bearing pads on the outside of castings or forgings in order to avoid completely machining the outside surfaces, the casting or forging wall between bosses not supported by mold contact must be thick enough to withstand the high internal curing pressures (regardless of what the internal curing pressures may be, the figure arbitrarily used for calculation purposes is 15,000 psi which represents a pressure greater than that involved in ordinary curing). Otherwise, provision must be made for relieving the pressure as the article is cured. In many cases such a treatment is not possible.

Rubber Stock: The use of rubber is different from the use of many other structural materials in that a rubber compound must be processed just prior to the fabrication of the rubber parts. The crude rubber must be mixed with a vulcanizing agent and various other ingredients whose natures and proportions depend on whether the cured rubber is to withstand abrasion, chill, compression, tension, shear, etc. There are numerous steps, many of them manual, in the

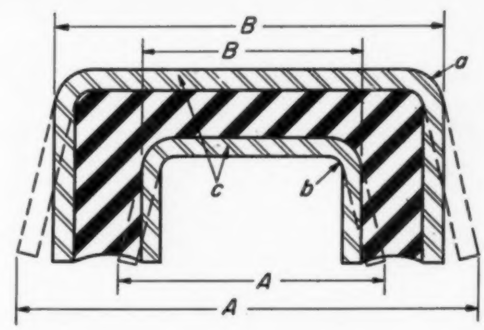
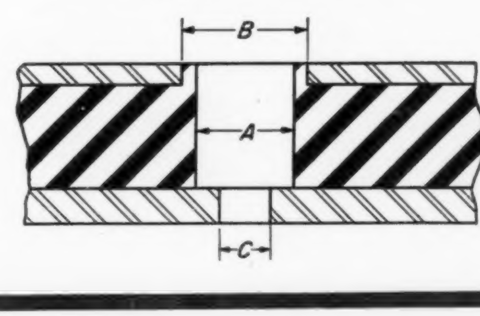


Fig. 3—Above—Radius *a* should be accurately machined or formed to fit the mold. Any variation in it should be plus. Radius *b* should be accurate. Any variation in it should be minus. The flat portions of the metals, *c*, should be reasonably free from irregularities which would prevent proper placing in the mold or would make it impossible to produce parts within required tolerances. If legs of channels spread as indicated by the dashed lines the dimensions *A* should not exceed the nominal widths specified. Dimensions *B* must always be less than corresponding dimensions *A*

Fig. 4—Below—Dimension *B* should be 1/32-inch larger than the desired nominal dimension *A*. Dimension *C* should be 1/32-inch greater than the diameter of the bolt or pin which passes through it. The tolerance on the holes in the metals should be minus 0.000, plus 0.032-inch



preparation of the unvulcanized rubber compound and in curing it. Possible variations in the crude rubber, in the manual operations of compounding, in mixing and forming, and in curing temperature, pressure and time are such that the cured rubber in "identical" parts may have Shore durometer hardness readings varying by as much as 10 points total. The use of rubber is also different from the use of many other structural materials, in that more consideration must

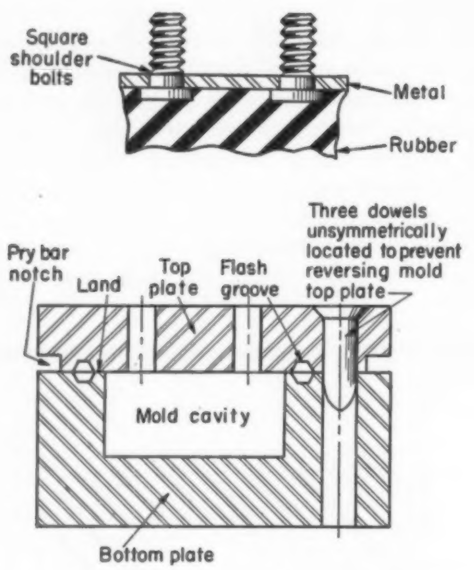


Fig. 5—Simple rubber and metal part and single-cavity, two-plate mold for production

be given to operating conditions, atmosphere, oils, temperatures, etc.

Molds: With few exceptions a mold is necessary for vulcanizing a structural rubber and metal part to the required shape. A mold consists of steel parts so disposed as to form a cavity, accurately machined to a specified size and shape, in which the rubber article is confined during vulcanization, *Fig.* 5. When the mold is subjected to curing temperatures of approximately 300 F, the mold, the metal parts and the rubber expand. The dimensions of the cavity must be such as to make allowances for all expansion effects; in particular, the cavity must be of such size that the rubber shrinkage which the cured sample experiences in cooling from approximately 300 F to room temperature is just sufficient to give the required dimensions to the rubber portion of the article. If parts are to be produced in volume, molds with multiple cavities, all exactly alike, should be used in order to reduce production costs.

A mold should be so designed that after vulcanization the rubber and metal part can be easily removed from the cavity without injuring the mold, the cavity, or the part itself. Therefore the proportions and shape of the part influence the mold design. Some molds are of two-plate design, as illustrated in *Fig.* 5, and others, due to the particular type of rubber part to be cured, necessitate the use of a three-plate mold, *Fig.* 6.

Curing Process: In order to determine production molding procedure, trial heats of parts have to be made experimentally after molds are received and before production is started. The trial heats determine the size and shape of the uncured rubber blank to be used. In many cases it is also necessary before starting production to determine correct curing conditions suitable for the size and shape of the article and to test the product to determine whether it has the specified physical properties.

The uncured rubber compound is a plastic with elastic properties but little, if any, tensile strength. The size and shape of the rubber blank for curing must be such as to permit loading and proper forming and "flow" in the cavity during cure. *Fig.* 6 shows a rubber blank and the metal parts prepared for curing in the mold cavity at the moment when the top plate is being put in place.

A platen press is used for vulcanizing. Live steam is circulated through the hollow press platens at a pressure approximately in the range 30 psi (274 F) to 115 psi (347 F), the particular pressure used being determined by the article being cured. The platens and molds are closed against the press head (or intermediate platens if a multiple platen press is used) by a hydraulic ram. Average press includes single rams from 12 to 30 inches in diameter. The hydraulic ram pressure is about 2000 to 2500 psi.

A ram of 20 inches diameter operated by a hydraulic pressure of 2000 psi exerts a total force of 628,000 pounds or 314 tons. The press used with such a ram would accommodate a mold about 24 to 32 inches square. This force of 314 tons closes the mold, presses the prepared uncured rubber blank into the cavity and squeezes out the excess rubber. Vulcanization starts immediately. Average rubber to metal adhesion parts require 25 to 40 minutes for a complete cure. Some exceptionally thick and large pieces

Fig. 6—Type of cured part which would require the use of a three-plate mold such as shown

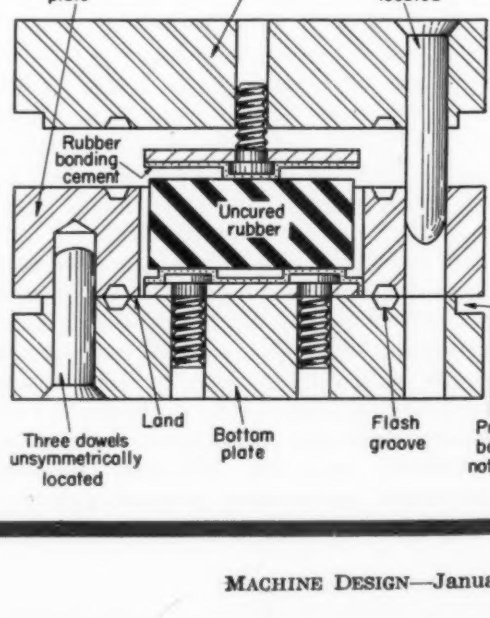
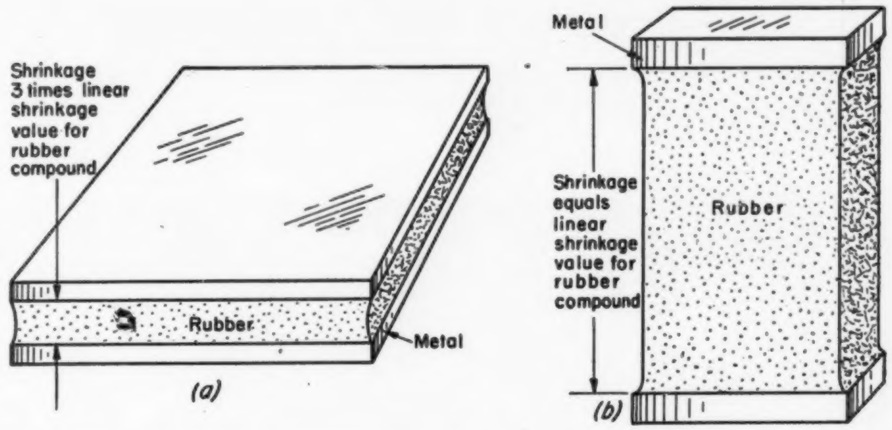


Fig. 7—Extreme conditions of shrinkage are illustrated by these adhered rubber parts. Design at *a* is the maximum because of large adhered surfaces while *b* is normal in shrinkage



such as are used for railroad and industrial applications require two to three hours for cure.

During cure, molding pressure is maintained by viscous resistance to flow of the rubber passing across the "land" which is the surface between the cavity and flash groove. Due to high internal curing pressures the rubber compound during vulcanization flows into all voids and openings in the cavity.

Tolerances on Cured Rubber Parts: Rubber overflow from the cavity is called "flash." The flash grooves serve as receptacles for the excess rubber flowing from the cavity. Where dimensional precision is required, the flash grooves are more than sufficiently large to accommodate all of the excess, thereby preventing the molds from being held open by rubber between the plates. The thickness of the flash is variable and is a factor which must be taken into consideration on finished thickness tolerances.

Typical commercial tolerances on dimensions across the parting line of precision-finished pieces made in a two-plate mold are as follows:

Dimension	Bonded Metal	No Metal
Less than ½-inch	±0.015-in.	±0.010-in.
½-inch to 1 inch	±0.020-in.	±0.010-in.
1 inch to 2 inches	±0.030-in.	±0.015-in.
2 inches to 3 inches	±0.040-in.	±0.020-in.

These tolerances are established on the basis of the metal parts being flat and relatively rigid. Variation from these conditions will be reflected in the cured part. Stocks of Shore durometer greater than 85 do not flow as easily during cure as do softer stocks, and therefore thicker flash must be expected for the harder stocks.

It is possible to check accurately the weight of each individual blank of uncured rubber compound prior to curing and thereby to reduce flash thickness to a minimum. Under this procedure the variation in part thickness due to flash thickness is considerably decreased but the cost of the part is materially increased due to the additional labor involved.

Shrinkage in Size: Shrinkage takes place as rubber cools from curing temperature to room temperature. The average linear shrinkage of cured rubber parts *not* bonded to metal is given in the following:

Shore Hardness	Linear Shrinkage (per cent)
30	2.20 ± 0.25
40	2.00 ± 0.25
50	1.75 ± 0.20
60	1.50 ± 0.20
70	1.40 ± 0.15
80	1.25 ± 0.15

Although induced shrinkage due to adhesion of rubber to metals is much too involved to be discussed in detail, a qualitative explanation of extreme conditions is in order. *Fig.* 7*a* illustrates a rubber sandwich in which the length and width of the rubber are large compared to its thickness. Since the rubber-metal bond prevents the length and width of the rubber in contact with the metals from shrinking, thermal stress relief results in a decrease in thickness. Volumetric shrinkage is approximately three times the normal linear shrinkage. For a 40-durometer stock, therefore, the shrinkage in thickness will be approximately 3 x 2 per cent or 6 per cent.

Fig. 7*b* illustrates a rubber sandwich in which the length and width of the rubber are both less than the thickness. Since the metal plates will affect the shrinkage only in the neighborhood of the rubber to metal contact, the shrinkage for such a sandwich can be considered as approximately that for a rubber part not adhered to metal faces. For a 40-durometer stock, therefore, the thickness will shrink 2 per cent and the length and width of the rubber at the midsection will both shrink 2 per cent.

Fig. 5 illustrates the type of shrinkage to be expected for a rubber part in which metal is bonded to only one side. When the rubber part, while still hot, is removed from the mold it has essentially the same shape as the mold cavity. Upon cooling to room temperature the sides shrink inward, and the non-adhered side ceases to be a plane surface.

Adequacy of Bond: In general, for high-quality rubber compounds bond strengths may run from 400 to 500 psi (averaged over bond interface area) for the type of mounting illustrated in cross section in *Fig.* 6. For cases of plane smooth surfaces with no bolt heads projecting into the rubber body and under the best manufacturing conditions the ultimate bond strengths may be twice as great.

Fig. 1—Micromotion filming the manual production assembly of retina film mounts recorded vital information which was subsequently used by the designer in planning an efficient machine to supplant the manual method

Fig. 2—Careful study of manual assembly methods led to conception of four different design ideas of possible machine construction. Precise

Micromotion Pictures . . .

IN DESIGNING or redesigning a machine for most efficient performance and convenient operation, a comprehensive survey should first be made of existing facilities and procedures. This is usually accomplished by making general observations of the devices being used, the manual operations performed, and by conducting extensive time and motion studies. Aside from other important considerations, if this particular phase is bypassed, or does not yield accurate and adequate information, the designer is placed under a distinct handicap.

More often than not, the man expected to design a machine for optimum efficiency is not able to visualize ideal relationships of operator, machine and workpiece. Under this circumstance, a realistic approach to the design problem is greatly hampered. Micromotion movies provide means of bridging the gap which sometimes exists between the designer and the design problem and they are being employed successfully in this respect. Case histories, as recorded by one company, prove that substantial returns are possible by utilization of this tool in the field of machine design.

Recognizing the designer's need for better understanding of basic requirements in the early stages of design, the industrial engineering department at Eastman Kodak Co. has recently begun using micromotion movies advantageously in the development and design of new machines. The approach being used is basically simple. It begins with the making of micromotion movies of the manual operation to be mechanized, or of the machine to be redesigned, as the case may be.

These motion pictures—which are taken at a constant speed of 1,000 frames per minute with a motor driven camera—enable determining precisely the exact length of time it takes to perform any given motion or series of motions. New operations are synthesized by using manual motions in various combinations with a special jig or improvised machine, and

time measurement of manual functions in 1/1000-minute by use of special projector established criterion for ultimate performance

Fig. 3—Designer and engineer check construction details of the new machine which now enables the assembly of film mounts in 75 per cent less time than by previous manual methods

. how they can aid in the design of machines

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an estimated cost of performing the operation by machine is determined. By comparing existing and estimated production costs, it is possible to determine whether or not further design effort is worthwhile. This procedure is particularly advantageous, considering today's critical shortage of designers.

Effectiveness of Approach

When micromotion photography is used, it is possible to supply machine designers with exact work place specifications, sizes, distances, most convenient position of controls and other data needed to design the best and most productive machine for doing the job while making the work much easier and less fatiguing for the operator. The effectiveness of this approach to machine design may be realized from the fact that the production on one Kodak job was increased almost 400 per cent when micromotion studies were made of the process and new machines designed for the work with the aid of facts obtained from the studies. In still another instance, micromotion studies of a method employed in a triple-assembly operation led to the design of a new machine which is much faster than previous hand methods.

In each of these instances the effectiveness of this new approach to machine design was enhanced by teamwork and co-operation of the production division, the engineering design division, and the industrial engineering division. Any one of these units could probably have surveyed the job alone, but the best designs were developed by the specialized units working together as a team. The manner in which such a study proceeds can probably best be illustrated by following the development of the triple-assembly device as it evolved from the application of these techniques.

Work began on the project when the production department asked for an evaluation of the existing method. An industrial engineer was assigned to the job and a movie was made of the manual assembly

operation by means of the micromotion camera, *Fig. 1*. The movies then were projected in a 16mm motion picture projector especially equipped for this work, *Fig. 2*. Such a projector has a frame counting device which allows the user to count the number of frames required for any given motion. This establishes, in thousandths of a minute, just how long each motion takes. A special reversing and single-frame projecting device, which enables the user to move the film through the projector at any speed in either direction, makes possible close study of details.

As the films were projected, the time required for each motion was charted for each hand. These charts showed at a glance how much useful work each hand was doing during the entire operation. Study of the micromotion film revealed that the left and the right hands of the operator had approximately equal amounts of work to do; both being comparatively busy. It became immediately obvious that to reduce the cycle length, as many manual operations as possible would have to be eliminated by specially designed fixtures or semiautomatic equipment.

From these observations, four basic methods of solving the problem were developed, plus a number of variations of each method. The four methods were carefully time-charted to show how efficient each might be. In working out these proposed techniques the engineer sketched in rough form his idea of how a machine might be constructed to perform certain functions that the operator had previously done by hand. The time it would take for the machine to do each operation was calculated using known factors and then integrated with the actual time required for unavoidable hand movements as revealed by the micromotion studies.

Selecting the Best Method

As a result of such preliminary studies the four proposed new methods indicated that savings of from 25 to 90 per cent could be made, depending upon the amount of money available for designing and constructing the machines and the complexity of the machines so built. To simplify the proposal study and make it easier to decide which design should be developed, a comparison chart was drawn which compared each proposed method on the most important points. This chart enabled management, at a glance, to see how each method "stacked up" alongside the others.

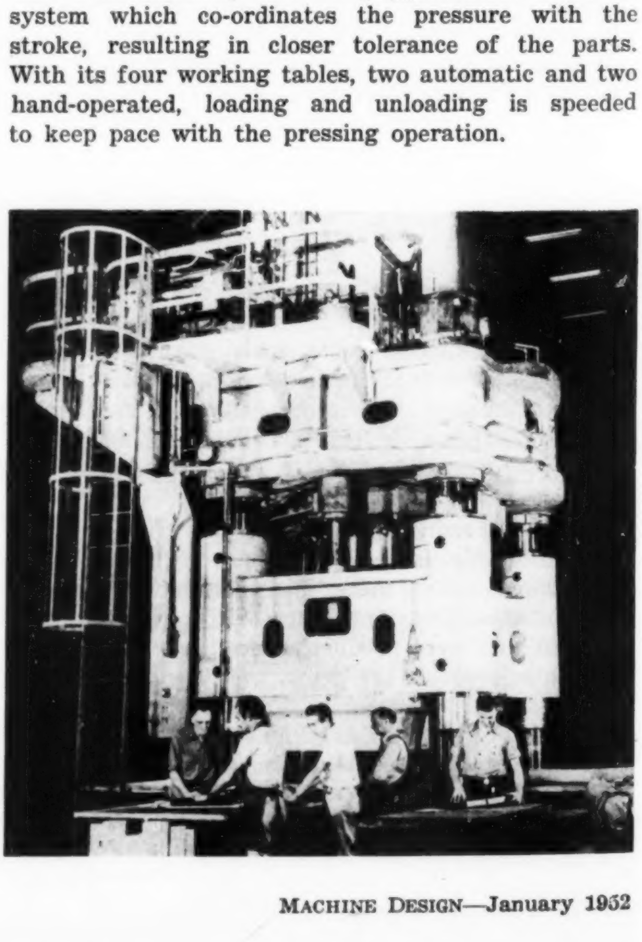
It may be of interest to note that the design idea selected and developed was not that which would enable the greatest savings, productionwise. A design giving slightly lower production savings was selected because the machine would be less complex and therefore, less likely to require a large initial investment in relation to potential return, and far more certain to operate smoothly with less maintenance.

The proposed method was subsequently accepted by management, and the basic ideas for the machine were then developed in detail by the machine designer. The final design as produced is illustrated in *Fig. 3*. When the new machine was placed in operation, a micromotion check revealed that a 75 per cent increase in production over that possible by the previous method had been realized. Greater convenience of control locations as established during the predesign study resulted in reduced operator fatigue, and fewer rejects occurred in production of the assemblies.

At the present time, Kodak engineers—none of whom are trained photographers—are shooting about 30,000 feet of film a year, to aid in machine design. To help estimate costs and time of proposed machine production operations, they also use standard elemental motion data published by the Methods Engineering Council. Experience has proved that they can expect production estimates to come within ± 4 per cent of the time standard determined by normal time study methods.

Giant Press for Jet Parts

LARGEST in the middle west, this 5000 ton hydraulic press, built for Boeing Airplane Co., by the Birdsboro Steel Foundry and Machine Co., weighs 876,000 pounds and is being used to form parts for the Boeing B-47 Stratojet, world's fastest jet bomber. Twice the capacity of the largest press used to make B-29 parts at Boeing-Wichita during the war, the new machine is necessary because of the size and thickness of the material processed in fabricating the B-47. Working from a single rubber die cushion, the press forms the metal around one or several single dies simultaneously. It is equipped with a control system which co-ordinates the pressure with the stroke, resulting in closer tolerance of the parts. With its four working tables, two automatic and two hand-operated, loading and unloading is speeded to keep pace with the pressing operation.



Charts Simplify Flywheel Calculations

By Harold M. Durham
Pittsburgh, Pa.

FLUCTUATING loads in a motor-driven system often suggest the use of a flywheel. As a renewable kinetic reservoir, a flywheel gives up energy to help the motor meet peak demands and regains it when the load is light. Generally, with a flywheel, a smaller, less expensive motor can be selected than that otherwise dictated by the peak load. Obviously, no flywheel is needed when the load is steady and none should be used if the system is reversing.

In this article, some of the fundamental aspects of design for flywheels will be briefly reviewed, and simplified calculation procedures will be presented for several types of problems that arise in the determination of flywheel proportions.

Flywheel Type: Because of manufacturing ease, higher safe speeds and uniformity of material, the disk type flywheel is preferred today for most applications, supplanting the older spoked-wheel heavy-rim designs of cast iron and cast steel.

A disk flywheel is usually low-carbon (0.15 to 0.25 per cent) steel, as rolled. Drilled and reamed for fitted bolts, the disk is generally assembled to a steel hub which, in turn, can be keyed to the shaft, *Fig.* 1.

Location, Coupling and Mounting: Various possible locations of flywheels are shown in *Fig.* 2. The flywheel mounting is best separated from the motor, which ordinarily is not designed to carry such an overhanging load and which, also, should be easily

Fig. 1—Overhung flywheel mounted on high-speed drive shaft

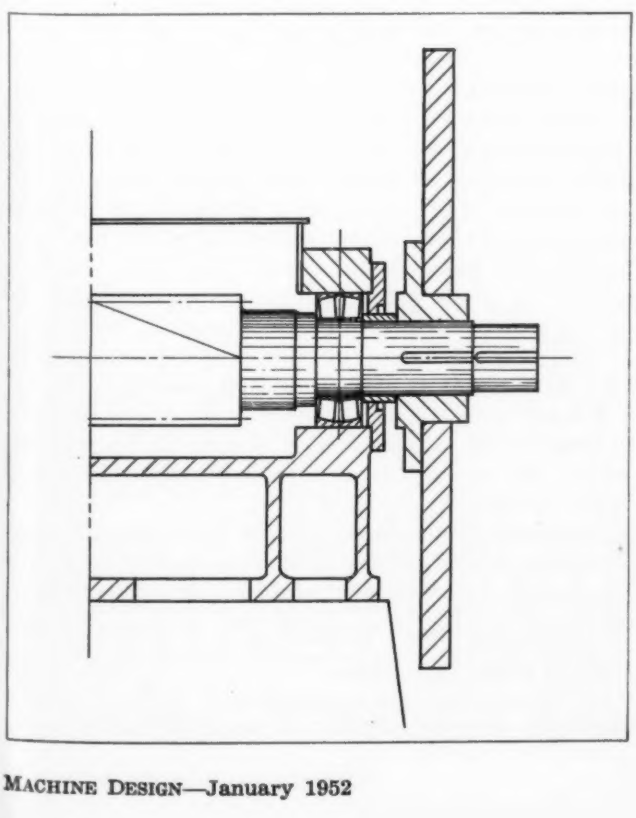
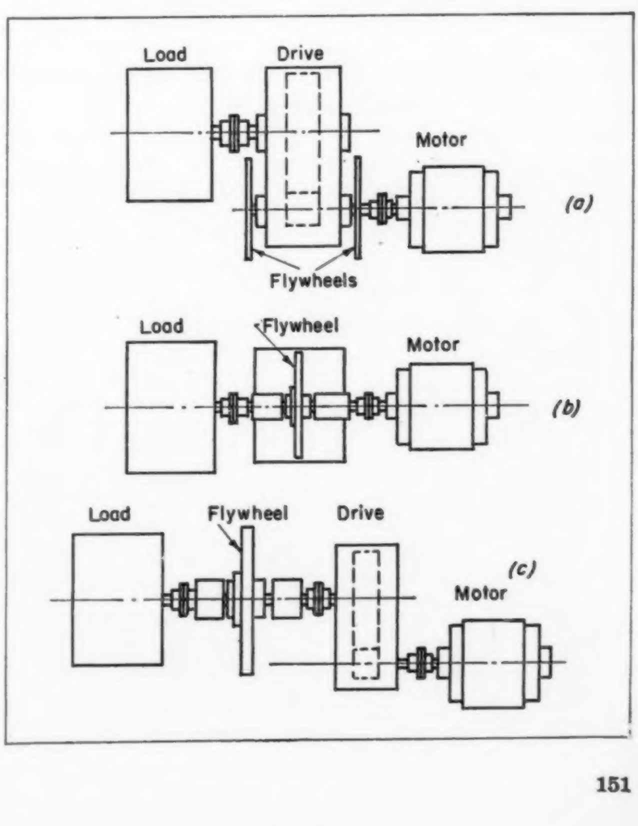


Fig. 2—Various possible locations of motor, flywheel, drive and load. Flywheel should preferably be located on high-speed shaft, such as at *a* or *b*



Flywheels

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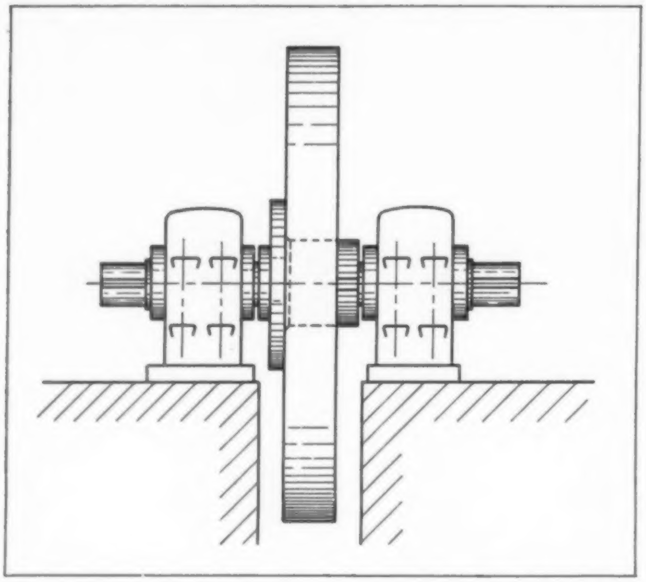


Fig. 3—Above—Typical low-speed shaft flywheel mounting

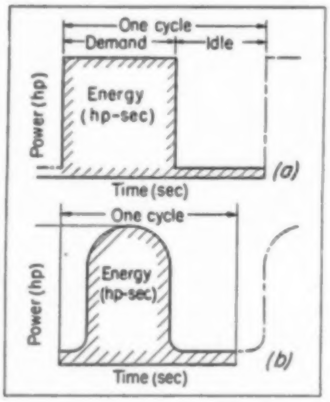
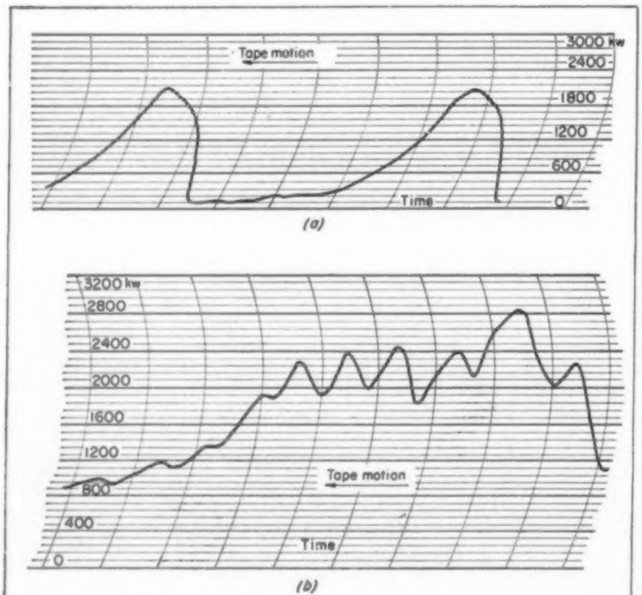


Fig. 4 — Left — Types of duty cycles. At *a* is shown the simplest form with constant demand and idle loads and, *b*, crank-press duty cycle

Fig. 5—Below—Actual duty cycle recordings of mill-drive installations



accessible for maintenance. Economy of space and material dictate that the flywheel be located on a high-speed shaft, as at ***Fig.*** *2a* or *b* in preference to *c*.

Flywheel shafts should be connected to others by means of flexible couplings instead of solid flanged couplings.

Large low-speed flywheels are sometimes mounted as shown in ***Fig.*** 3. The supporting shaft is carried on two pillow blocks, one of which restrains the unit against axial motion. This arrangement necessitates special care to insure alignment. It is preferable to locate the flywheel on the high-speed shaft on an integral machine bedplate, connected to motor and load as shown in ***Fig.*** *2b*. Where the flywheel is mounted overhung on the drive as in ***Fig.*** 1, the amount of overhang should be a minimum, and the shaft designed for minimum deflection and few shoulders.

Duty Cycle: When need for a flywheel is indicated, the first requisite for a study of the problem is a diagram showing the duration, frequency and extent of the loads. This diagram is the *duty cycle* of the system being studied. With power and time as co-ordinates, areas of this diagram represent energy and apply to any shaft in the system.

In ***Fig.*** *4a* is shown a simple type of duty cycle where the power required is uniform over the demand period as well as over the idle period. Simple rectangular elements of area represent energy for this case. Durations of periods at demand and idling are as important as the intensity of the loads, for both affect the sizes of motor and flywheel.

With a duty cycle of the type shown in ***Fig.*** *4a*, for a motor application without a flywheel, an analysis should dictate a motor capable of carrying the maximum load, of course. At the same time, however, the motor should be able to withstand the heating predicted by a root-mean-square analysis of the load and idling periods.

At this particular stage of the analysis possible use of a flywheel should enter the picture. When the lengths of the idle time periods are sufficient, a flywheel can be employed and a smaller, less costly motor selected.

Areas of the duty cycle periods are used in various computations and must be evaluated, either from tentative diagrams or from actual graphs from recording meters. ***Fig.*** *4b* shows a diagram for a crank press; ***Fig.*** 5 shows actual records from two steel-mill drive-motor installations. Depending upon the load conditions, areas under curves can be integrated by one of several methods: by mathematical integration of a known function, by graphical approximations such as Simpson's rule, or by a planimeter.

Energy required by the duty cycle is one of three factors to be combined in a successful design; the other two are flywheel energy characteristics and motor behavior.

Flywheel Proportions: From fundamental equations, the following expression for energy of a steel flywheel in the form of a simple disk can readily be derived:

$$E_w = N^2 d^4 h \; (6 \times 10^{-11}) \quad \text{(1)}$$

where E_w = energy, hp-sec; N = speed, rpm; d = di-

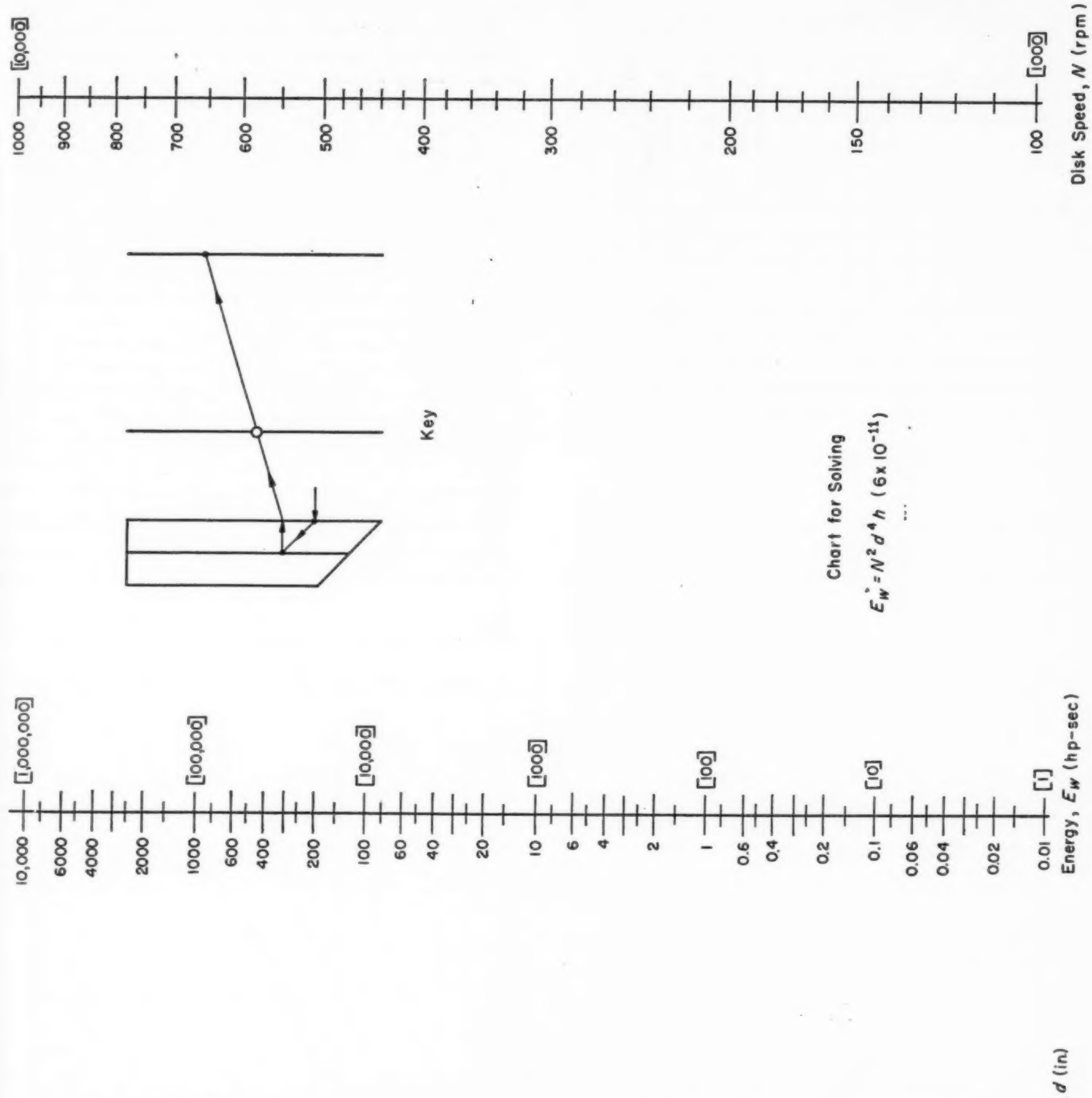


Fig. 6—Nomogram for determining stored energy in horsepower-second units for steel disk flywheels. For higher speeds, use bracketed N range and read bracketed scale for E_w

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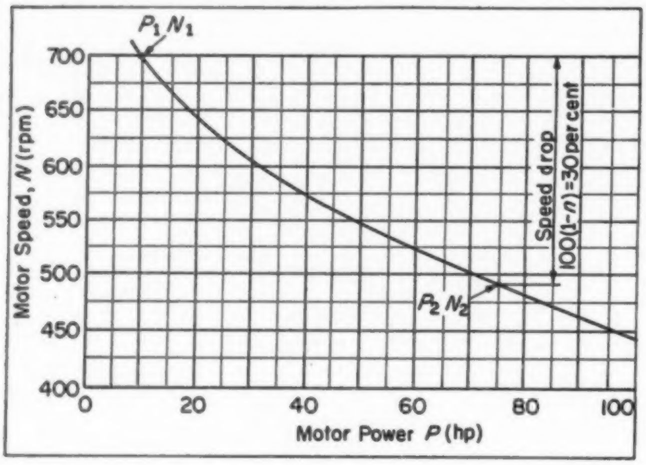
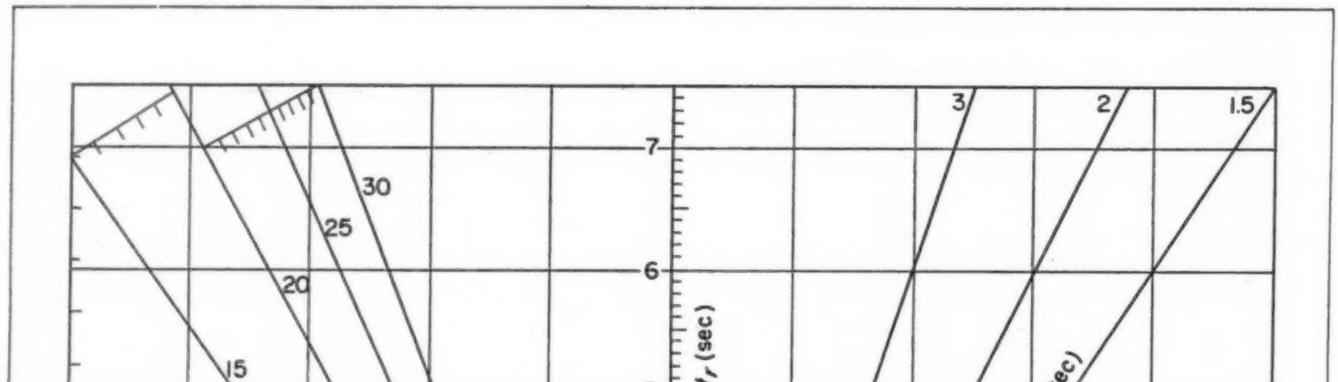


Fig. 7 —Above—Speed characteristic curve of direct-current compound motor

Fig. 8—Below—Chart for solution of direct-current motor and flywheel problems



ameter, inches; and h = thickness, inches. Expressing energy in horsepower-seconds is simply a convenience in general calculations (1 hp-sec = 550 ft-lb).

The flywheel energy equation is shown as a nomogram in *Fig.* 6. Energy of a flywheel at a particular speed can be rapidly computed by use of the nomogram. For example, find E_w for d = 78 inches, h = 3 inches, and N = 600 rpm. Entering the grid at d = 78, proceed parallel to the diagonals until intersecting the vertical, h = 3. Return on the horizontal to the right-hand limit of the grid. From this latter point, project a line through the 600 point on the N scale. Read E_w = 2400 hp-sec on the central scale. With E_w and N known, proper companion values of d and h can be found easily by inversion of the nomographic sequence.

Motor Characteristics: Drop in speed for a particular motor depends not only on the load but also on the type of motor. Characteristics are different for alternating and direct current, and vary also with the windings and service for which the motor is designed. A typical power-speed curve of a direct-current compound wound motor is shown in *Fig.* 7. The curve of a shunt motor has a flat characteristic, mak-

ing it undesirable for flywheel applications.

Characteristic curves should be available when a motor for a flywheel drive is being considered. Motor manufacturers should be consulted for this information; assumption should be avoided. With compound-wound motors, it is well to remember that the load-speed behavior depends upon the percentage of compounding built into the particular motor type. Some compound motors have a rising speed characteristic.

Direct-Current Motor Drive: A chart which facilitates solution of direct-current motor and flywheel problems is presented in *Fig.* 8. The following nomenclature is involved.

E_w = Flywheel energy, hp-sec
ΔE = Energy delivered by flywheel in losing speed by $100(1-n)$ per cent, hp-sec
N_1 = Highest motor speed (for idle load), rpm
N_2 = Lowest motor speed, rpm
n = Flywheel and motor speed fraction (N_2/N_1)
P_a = Average effective motor power, hp
P_L = Power required by load, hp
P_1 = Motor power at idle load speed (at N_1), hp
P_2 = Motor power at lowest speed (at N_2), hp
t = Load period, sec
t_r = Recovery period, sec

These factors are linked by several simple relationships. Average effective motor power is the mean between the idle load power and maximum power:

$$P_a = \frac{P_1 + P_2}{2} - P_1 = \frac{P_2 - P_1}{2} \qquad (2)$$

Energy transferred by the flywheel during a drop in speed is

$$\Delta E = t\,(P_L - P_a) = E_w\,(1 - n^2) \qquad (3)$$

The following expressions, derived from Equations 2 and 3, are useful for the solution of typical problems:

$$\frac{\Delta E}{P_a} = t\left(\frac{P_L - P_a}{Pa}\right) = t\left(\frac{P_L}{P_a} - 1\right) \qquad (4)$$

$$\frac{E_w}{P_a} = t\left(\frac{P_L}{P_a} - 1\right)\left(\frac{1}{1-n^2}\right) \qquad (5)$$

$$E_w = t\,P_a\left(\frac{P_L}{P_a} - 1\right)\left(\frac{1}{1-n^2}\right) \qquad (6)$$

$$t_r = \frac{E_w\,(1-n^2)}{P_a} = \frac{\Delta E}{P_a} \qquad (7)$$

Use of the equations and the chart is best demonstrated by an example. Determine the energy capacity of the flywheel needed for the duty cycle shown in *Fig.* 9 and the recovery period. Assume that a motor having the characteristics plotted in *Fig.* 7 will be used.

With the speed drop limited to 30 per cent, motor

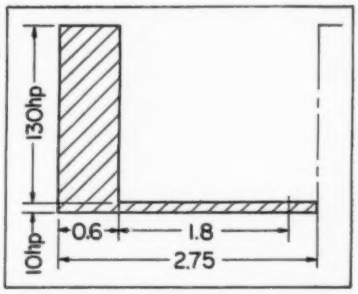
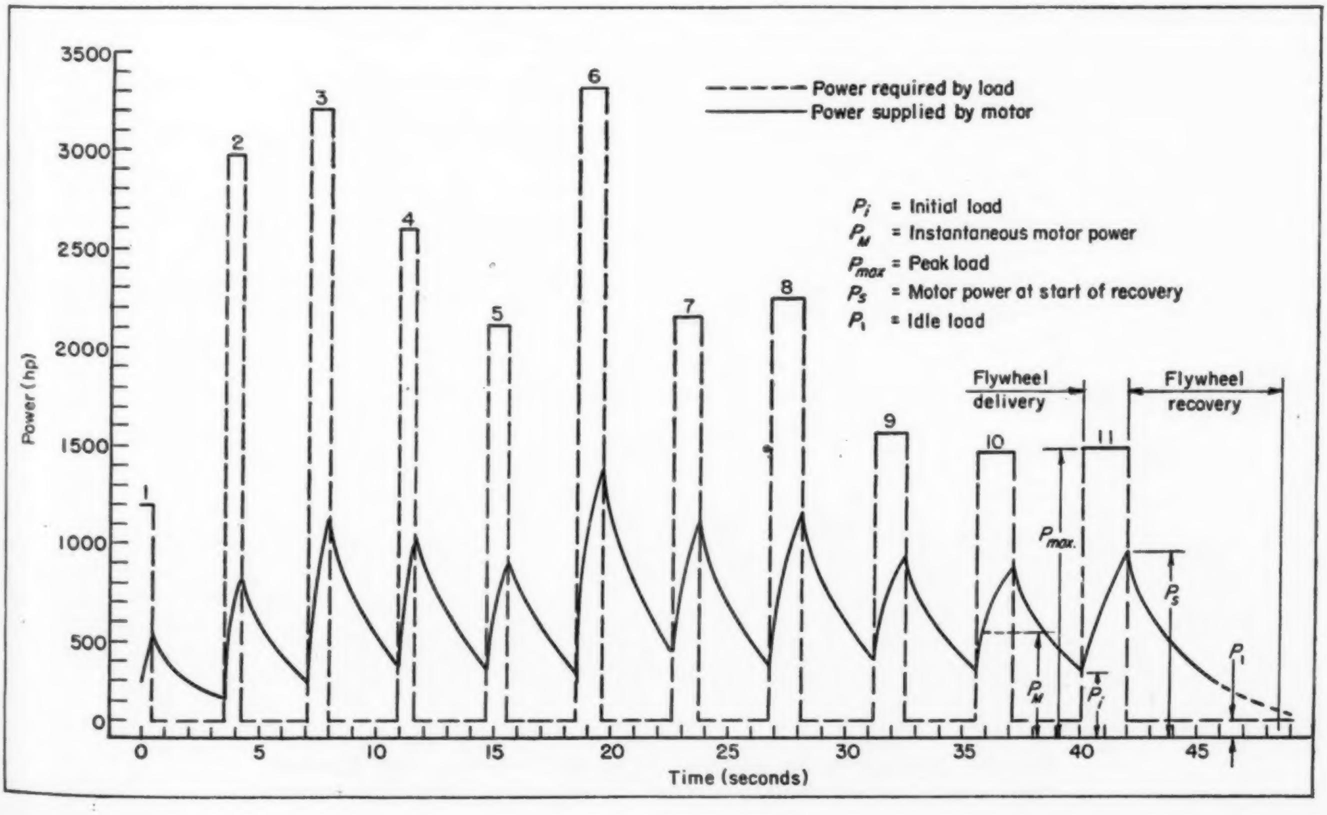


Fig. 9—Duty cycle for typical problem also involving the motor characteristic curve of *Fig.* 7

Fig. 10—Below—Flywheel and motor load studies for wound-rotor motors showing variations in initial and final load values through a number of periods



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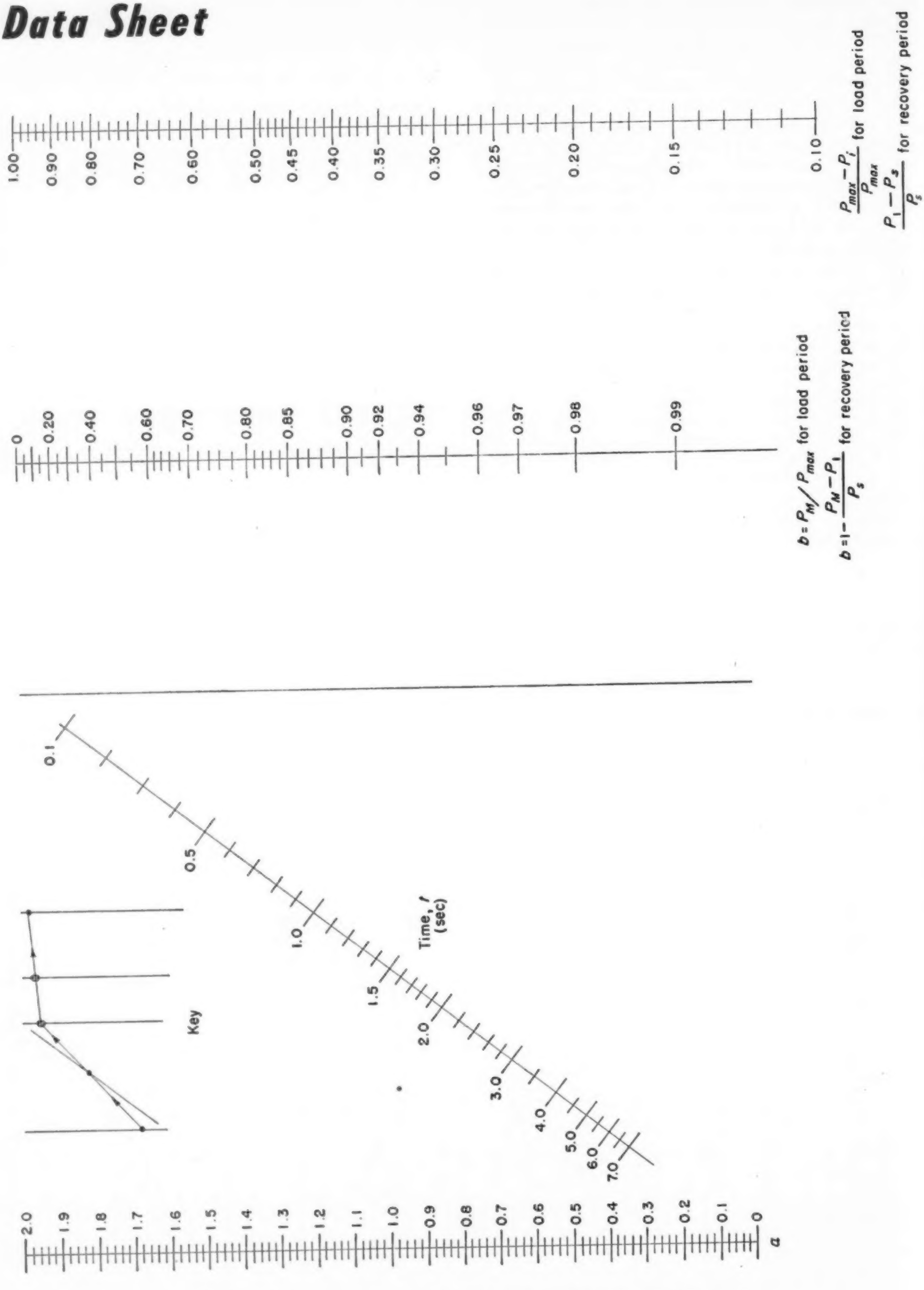


Fig. 11—Nomogram for solution of problems dealing with flywheel applications to alternating-current wound-rotor induction motors

and duty data, *Figs.* 7 and 9, provide the following: For idle load, $P_1 = 10$ hp, $N_1 = 700$ rpm; for speed drop of 30 per cent, $N_2 = 490$ rpm, $P_2 = 75$ hp.

From Equation 2, average effective power, $P_a = (75-10)/2 = 32.5$ hp. Next, from *Fig.* 9, power required by load, $P_L = 130\ (1) = 130$ hp-sec. Then, by calculation, $(P_L/P_a) - 1 = (130/32.5) - 1 = 3$. Now, three of the five variables in the chart of *Fig.* 8 are known: $(P_L/P_a) - 1 = 3$; from the duty cycle, *Fig.* 9, load period $t = 0.6$ sec; and by limiting premise, speed drop $100(1-n) = 30$ per cent. Letting these three factors determine the path of the marked solution in *Fig.* 8 yields $E_w/P_a = 3.5$ and recovery period $t_r = 1.8$ sec. Then $E_w = 3.5\ P_a = 3.5(32.5) = 114$ hp-sec.

Actual dimensions of a disk flywheel that will store 114 hp-sec can be determined from *Fig.* 6. Speed depends upon location of the wheel in the system. If the flywheel is so located as to turn at the motor speed, its dimensions can be found from *Fig.* 6 by projecting a line through $N = 700$ and $E_w = 114$ to the d scale. Diameter d varies from, say, 44½ inches for thickness $h = 1$ inch to 34 inches for $h = 3$. Selection of specific proportions in any design may depend upon economy of material and weight, space limitations, or safe peripheral velocity.

In *Fig.* 8, if required values of recovery period t_r and E_w/P_a are beyond their respective scales, they may be found by an easy method. Divide the actual load period t by any convenient factor that places the results within scale range. Read the reduced values of t_r and E_w/P_a and simply multiply them by the selected factor to obtain the actual values.

Alternating-Current Motor Drive. On the assumption that the power output of an induction motor is proportional to the drop in speed below the no-load speed, the following relationship* can be developed for the instantaneous motor load during the load period:

$$P_M = P_{max} - \frac{P_{max} - P_i}{e^{at}} \qquad (8)$$

Similarly, instantaneous motor load during the recovery period is

$$P_M = P_I - \frac{P_1 - P_s}{e^{at}} \qquad (9)$$

*R. G. Isaacs—"Application to the Electric Drive," an addendum to "Driving Sheet and Tin Plate Mills" by J. S. Caswell; *Blast Furnace and Steel Plant*, Dec., 1930, pages 1833 and 1835.

In these equations,

$$a = 0.5\,\frac{P}{E_w s} \qquad (10)$$

and additional nomenclature is as follows:

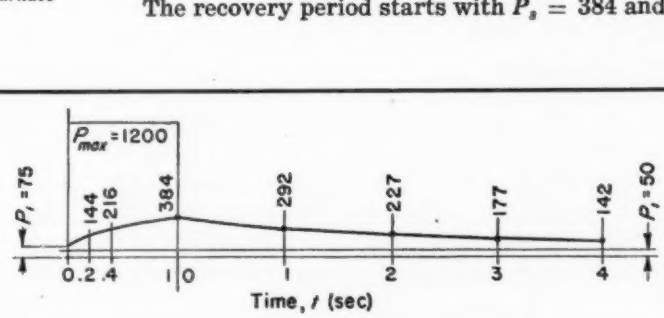
E_w = Flywheel energy, hp-sec
P = Nominal power of motor, hp
P_i = Power required by initial load, hp
P_M = Instantaneous motor power, hp
P_{max} = Power required by maximum load, hp
P_s = Motor power at start of recovery, hp
P_1 = Power required by idle load, hp
s = Slip, reduction in speed of flywheel, expressed as a decimal
t = Time, sec

By Equations 8 and 9, curves of instantaneous motor load can be calculated. Such curves, shown in *Fig.* 10 superimposed upon a duty cycle, are calculated by introduction of consecutive time increments and use of total elapsed time within a given load or recovery period for t in Equations 8 and 9. The nomogram in *Fig.* 11 is based on this equation and can be used for rapidly determining motor power curves.

Use of the chart in *Fig.* 11 is easily demonstrated by the solution of a typical problem. Assume the following conditions: motor $P = 250$ hp; flywheel energy $E_w = 3900$ hp-sec; slip $s = 10$ per cent; maximum load $P_{max} = 1200$ hp applied for 1 sec; initial load $P_i = 75$ hp and idle load $P_1 = 50$ hp. Plot the motor power curve over the load period, when the flywheel is giving up energy, and the recovery period, when the flywheel is regaining energy. Initial load P_i will be used for the load period calculation, idle load P_1 for the recovery.

First, from Equation 9, $a = 0.5P/E_w s = (0.5)(250)\ /\ (3900)\ (0.10) = 0.32$. Also, calculate: $(P_{max}-P_i)/P_{max} = (1200-75)/1200 = 0.94$. Then, enter *Fig.* 11 with $a = 0.32$, project through $t = 1$, and strike the ungraduated scale. From this intersection project a line to $(P_{max}-P_i)/P_{max} = 0.94$ and read $b = P_M/P_{max} = 0.32$. Then, by substitution of $P_{max} = 1200$, $P_M = 0.32\ (1200) = 384$ hp. The same procedure can be used for any fractional parts of the load period, such as $t = 0.2$, 0.4, etc. Results are tabulated and plotted in *Fig.* 12.

The recovery period starts with $P_s = 384$ and $t = 0$;



Time, t	b	$bP_{max} = P_M$
0.2	0.12	144
0.4	0.18	216
1.0	0.32	384

Time, t	b	$P_1 + P_s(1-b) = P_M$
1	0.37	50 + 242 = 292
2	0.54	50 + 177 = 227
3	0.67	50 + 127 = 177
4	0.76	50 + 92 = 142

Fig. 12 — Tabulations of calculated factors and development of motor power curve during load and recovery periods

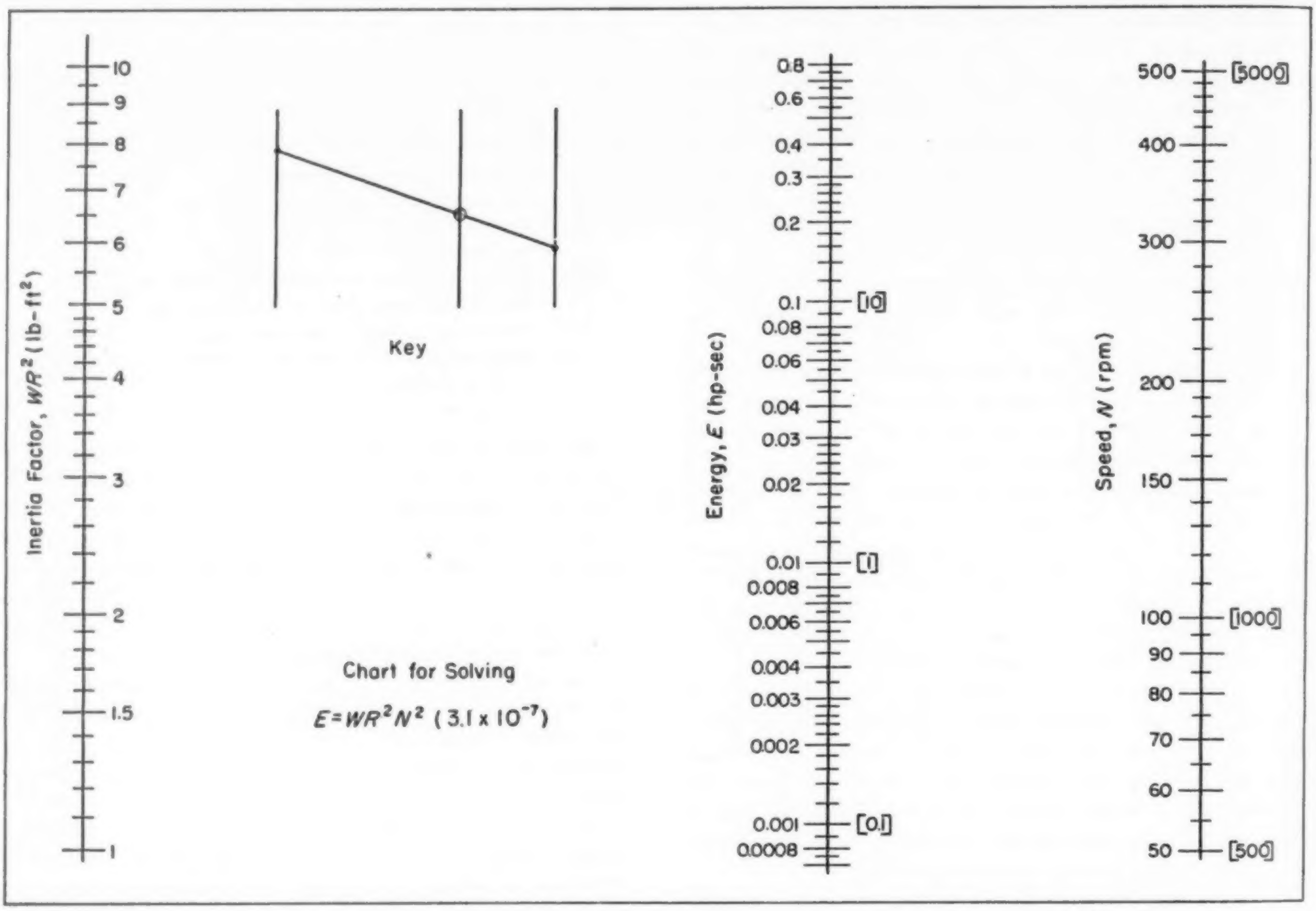


Fig. 13—Nomogram for determining energy E in horsepower-second units from known WR^2 and speed N

at $t = 1$, the procedure is to project from $a = 0.32$ through $t = 1$ to the ungraduated scale. Then, join this point with $(P_1 - P_s)/P_s = (50 - 384)/384 = -0.87$, ignoring the minus sign. Then, on scale b, read 0.37. As shown in the recovery tabulation of *Fig.* 12, $P_M = P_1 + P_s\ (1 - b) = 50 + 384\ (1 - 0.37) = 292$ hp. This value is shown plotted on the curve of *Fig.* 12. Additional values can be obtained by using $t = 2, 3$, etc., until the ordinates reach the idle load ($P_1 = 50$ hp).

With the nomograph of *Fig.* 11 quick trials can be made of various flywheel sizes and slip values with resultant motor loads.

Frequently, the cycles vary with respect to peak loads, period, and number of periods as depicted in *Fig.* 10. Thus, study of complete periods forming a group is necessary. Often a flywheel does not fully recover its energy loss during the available recovery period. This circumstance accounts for the variety of initial load values P_i applying to each period of energy delivery.

The previous calculations have been intentionally simplified by consideration of the flywheel energy only. For a complete analysis of such a system, energy of all components must be included. The inertia moment of such parts as motor armature couplings, drive parts and the machine proper may be converted to horsepower-second units by the following conversion:

$$E = WR^2 N^2\ (3.1 \times 10^{-7}) \qquad (11)$$

where E = energy, hp-sec; WR^2 = inertia moment, lb-ft^2; and N = rpm.

A nomogram for the rapid solution of Equation 11 is presented in *Fig.* 13. It will be observed that the chart range of WR^2 is 1 to 10. Since E varies in direct proportion with WR^2, the decimal point of the known WR^2 can be adjusted to bring the value into scale range. The resulting value of E can be adjusted inversely by the same factor. Range of speed N is extended simply by the alternate bracketed scale. It must be used in conjunction with the bracketed scale for E.

As an example, find E for $WR^2 = 44$ and $N = 900$ rpm. Project a straight line from $WR^2 = 44/10 = 4.4$ to $N = [900]$. Read $E = [1.1]$, and correct by factor of 10: $E = 1.1\ (10) = 11$ hp-sec.

DESIGN ABSTRACTS

Flat-Wound Tension Springs

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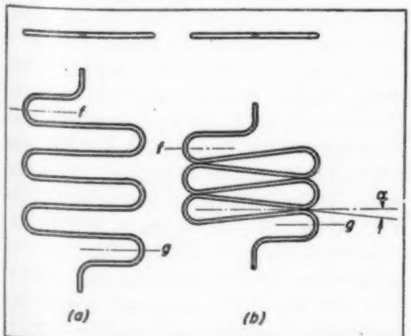
DESIGN of spring-actuated mechanisms often can be improved by the use of a flat-wound spring. *Fig.* 1 illustrates the physical appearance of flat-wound springs. This type of spring permits its utilization in many places where the conventional helical spring would be difficult to apply. In many mechanisms, the narrow cross section of the flat-wound spring will result in a considerably more compact design. In addition, it will permit the line of action of the spring to be located closer to the bearings, thus reducing the cantilever load on the mechanism. This is especially true in certain over-center, snap-action devices.

The objective of this paper is to provide the designer with easily applied formulas and a nomogram which will quickly and easily permit him to determine the applicability of the flat-wound spring to a design problem.

Fig. 1—Flat-wound springs shown in unloaded state



While a flat-wound spring of the type shown in *Fig. 1a,* may be used in compression, it is susceptible to buckling in the plane of the spring. Because of this instability, it is therefore desirable to guide the moving end and to restrain the sides of the spring when used in compression.

Complete deflection and stress equations (derived in the original paper) are given in the accompanying table. The nomogram, *Fig.* 2, was constructed from the simplified equation for deflection per coil ($\alpha = 0$) and the actual equations for maximum stress in round and square wires.

Use of Nomogram: To find the diameter d of wire required for a flat-wound spring, having y inches deflection per coil, with an allowable stress S under a load of P pounds, use the following procedure:

On the PS scale, locate the value of PS (e.g., $P = 20$ lb and $S = 50{,}000$ psi; then $PS = 1{,}000{,}000$). On the y-scale, locate the required value of the deflection per coil in inches (e.g., $y = 0.01$ in.). Connecting these two points with a straight line, locate the intersection of this line and the Q-scale. From this intersection, project horizontally to intercept one of the curves of the upper group. The choice of this curve will depend upon the ratio of the bend radius to the diameter of wire which is assumed and whether square or round wire is to be used (e.g., if round wire with $R = 2d$ is selected, use curve 6). At this point, the W/R ratio may be read on the horizontal W/R scale (e.g., $W/R = 2.7$). From the point of intersection on the upper curve, project vertically to a lower curve having the same letter and the suffix A (e.g., curve 6A). From this point, project horizontally until once again at the Q-scale. Now, locate the computed value of P/S (e.g., $P/S = 4 \times 10^{-4}$) and connect this point and the last intersection on the Q-scale with a straight line.

The intersection of this line and the d-scale gives the required value of d (e.g., $d = 0.15$ in.). For square wire, d = distance across flats.

Any of the scales of the nomogram may be expanded in the following manner: The PS scale may be multiplied (or divided) by 100 if, at the same time, the y-scale is multiplied (or divided) by 10. Similarly, the P/S scale may be multiplied (or divided) by 100 if the d-scale is multiplied (or divided) by 10. Note that while the PS and y-scales must be expanded simultaneously, and the P/S and d-scales must be expanded simultaneously, it is not necessary to expand all four scales at once.

The nomogram is constructed for steel ($E = 30{,}000{,}000$ psi); however, it may be modified for other materials by dividing the values on the y-scale by the modulus of elasticity

Nomenclature

- d = Diameter of round wire, or side of square wire, in.
- E = Modulus of elasticity of wire material, psi
- I = Moment of inertia of wire cross section, in.^{4}
- K = Ratio of width of spring to radius of bend, in. per in.
- P = Total load on spring, lb
- R = Mean radius of bend of spring, in.
- S = Total maximum stress, psi
- W = Width of spring, center to center of bend radius, in.
- y = Deflection per coil, in.
- α = Initial angle of spring bend (constant), radian

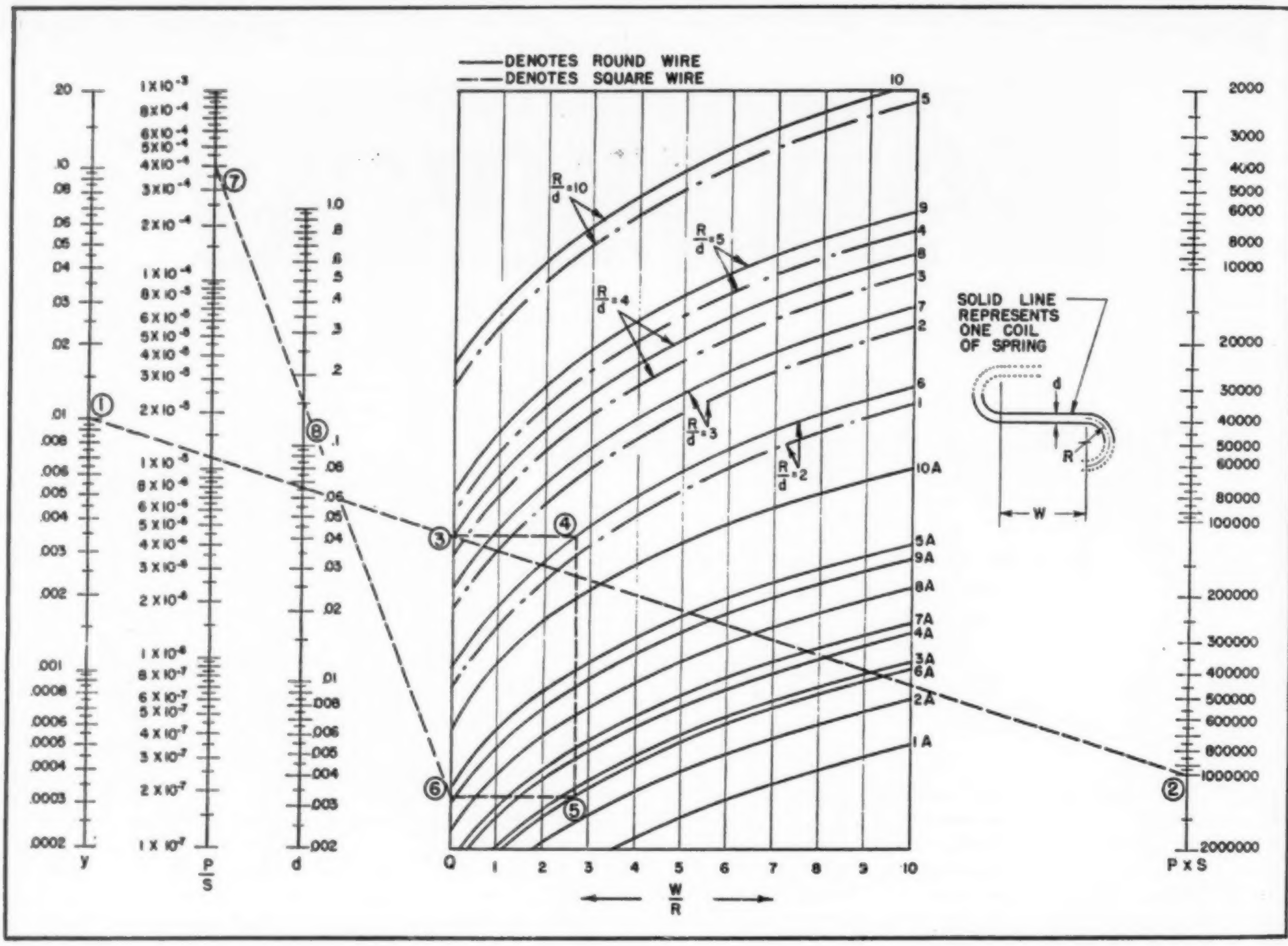


Fig. 2 — Nomogram for determining proportions of flat-wound springs

Equations for Flat-Wound Tension Springs

Actual equation for deflection per coil

$$y = \frac{PR^3}{12\,EI}\left[\frac{K^3}{\cos\alpha} + 3K^2(\pi + 2\alpha - 2\tan\alpha) + 12K\,(\sin\alpha\tan\alpha + 2\cos\alpha) + 6\left(\pi + 2\alpha - 2\sin\alpha\cos\alpha - \frac{4}{3}\sin^2\alpha\tan\alpha\right)\right]$$

Simplified equation for deflection per coil ($\alpha = 0$)

$$y = \frac{PR^3}{12\,EI}\,(K^3 + 9.43\,K^2 + 24\,K + 18.85)$$

Actual equation for maximum stress in round wire

$$S = P\left[\frac{\left(10.19\frac{R}{d} + 5.09\,K\frac{R}{d}\right)\left(1 + \dfrac{1}{\frac{10}{3}\frac{R}{d} - \frac{5}{3}}\right) + 1.28}{d^2}\right]$$

Actual equation for maximum stress in square wire

$$S = P\left[\frac{\left(6\frac{R}{d} + 3\,K\frac{R}{d}\right)\left(1 + \dfrac{1}{4\frac{R}{d} - 2}\right) + 1}{d^2}\right]$$

of the material, and multiplying by 30,000,000.

EXAMPLE 1: Suppose $P = 40$ lb and $S = 100{,}000$ psi ($PS = 4{,}000{,}000$), and $y = 0.06$ in. Expand the PS scale by multiplying by 100, at the same time expanding the y-scale by multiplying by 10. It is not necessary to expand the P/S and d-scales.

EXAMPLE 2: Suppose $P = 0.001$ lb and $S = 200{,}000$ psi ($P/S = 5 \times 10^{-9}$). Expand the P/S scale by dividing by 100 and, at the same time, divide the d-scale by 10. Note that the PS and y-scale need not be changed.

EXAMPLE 3: Suppose $P = 500$ lb and $S = 200{,}000$ psi ($PS = 100{,}000{,}000$; and $P/S = 2.5 \times 10^{-3}$), $y = 0.9$ in., and $R = 5d$. Expand the PS scale by multiplying by 100, and the y-scale by multiplying by 10. Locate 0.9 on the expanded y-scale and 100,000,000 on the expanded PS scale. Also, it is necessary to expand the P/S scale by multiplying by 100, at the same time multiplying the d-scale by 10. Now, locating 2.5×10^{-3} on the P/S scale and using the foregoing method of solution, it is

found that $W/R = 2.1$ and that $d = 0.55$ in. on the expanded d-scale. In this particular example, it was found necessary to expand all four scales.

Note that when interpolating for values of R/d, other than those given on the nomogram, interpolate between successively numbered curves for the particular cross section in question (round or square). This applies to both the upper and lower sets of curves on the nomogram.

Experimental Results: The derived formulas indicate that flat-wound springs have a constant spring rate, and the experimental results confirm this condition. Since K changes slightly during the deflection of the spring, there is theoretically a slight variation in spring rate which may be neglected in the majority of applications, it being less than the experimental error.

The deflection equation was verified by measuring the deflections of a number of flat-wound springs of varying wire sizes and dimensions. All of the springs tested were made from oil-tempered spring wire.

Fig. 3 shows a photoelastic model which was used to determine the stress distribution in the spring. The model was made of photoelastic plastic having the following dimensions: Depth of section, 0.500 in.; width of section, 0.500 in.; W/R ratio, 3.2; R/d ratio, 2.5; and R, 1.25 in. A 5-lb load on the model produced a deflection of 0.345 in. with the stress pattern shown in *Fig.* 3. The illustration shows a greater line density and, therefore, a higher stress level near the inner radius of the bend than at the outer radius. This is the result of two factors. One is the shift in the neutral axis due to the curved-beam effect. The other is the straight tension stress, which adds to the bending stress on the inner surface, and subtracts from the bending stress on the outside surface.

It was found that for most flat-wound spring applications the spring ends could be considered inactive for purposes of calculation without serious error. Thus, for all practical purposes, the number of coils may be determined by counting the number of full free bodies between the points marked f and g in Fig. 1 (five coils shown). For ends, as shown in *Fig.* 1, the maximum stress will always occur in the spring proper rather than the ends.

Conclusions: The flat-wound spring has, for all practical purposes, a constant spring rate and, therefore, can be used in much the same way as a helical spring.

The list of formulas derived in this paper may be used to solve mathematically the stress and deflection values for flat-wound springs.

The formulas and nomogram have been checked experimentally and found to be correct within 6 per cent.

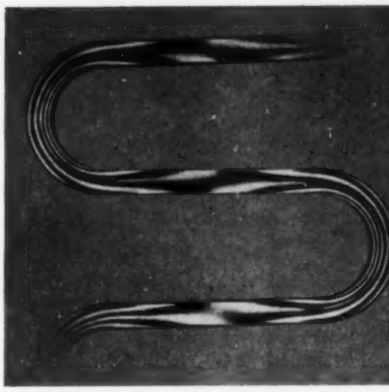


Fig. 3—Photoelastic model showing stress distribution in a flat-wound spring

The value of α, the original angle of spring bend, has little effect on deflection, and the use of the simplified expression for deflection may be used with small error.

Consideration of the use of flat-wound springs should be given where spring space limitations are severe or where spring forces result in excessive cantilever loads on bearings.

From Paper No. 51—A-59, "Design of Flat-Wound Tension Springs," presented at the ASME Annual Meeting in Atlantic City, N. J., November 25-30, 1951. Complete copies may be obtained from ASME, 29 W. 39th St., New York 18; $0.25 each to members, $0.50 to nonmembers.

Boron Engineering Steels

By Charles M. Parker
Metallurgical Engineer
American Iron and Steel Institute
New York, N. Y.

AT THE start of World War II we were faced with a shortage of alloying elements which forced the development of the lean nickel-chromium-molybdenum engineering steels known then as National Emergency Steels. For some time after their announcement their use was made mandatory by the War Production Board. Many consumers used them reluctantly and their attitude toward the steels was not improved by the terms "emergency" and "substitute" which were so freely used to describe them.

In addition, conventional mechanical property data on which some engineers relied were not immediately available and the implications of the standard end quench hardenability test were little understood or appreciated outside the steel and automotive industries. There was also some unwillingness to accept and use data obtained from experimental electric furnace heats and even from single heats of commercial size.

As time wore on and as production and use of the new steels mounted into the millions of tons, animosity toward the new steels diminished and men who had previously sought to find faults started to look for virtues. That they found them is amply demonstrated by the fact that after the war most of the "emergency" steels became standards and their peacetime production has been in the neighborhood of 15 per cent of total engineering alloy steel.

While that development work was going on a search was under way for new alloying elements and new methods of obtaining an "alloy effect" in both plain carbon steels and leanly alloyed steels. Some experiments with complex deoxidizing agents containing vanadium, titanium and aluminum showed considerable promise in improving the hardenability of carbon and engineering alloy steels. Moreover, heat treated in the same section size to the same hardness the steels deoxidized with the complex alloy showed marked increases over conventional steels of the same base composition in tensile strength, yield point, elongation, reduction of area, and impact strength.

At about the same time other experiments were conducted using fer-

roboron, and the same results were obtained as with the complex deoxidizers, which at the time were not known to have contained boron.

The quantity of boron which is effective in improving the hardenability of steel is almost infinitesimal. For that reason it was considered "smart" during the development days to characterize boron steels by such colorful names as "vitamin treated," "needled," "alloy treated," or "hopped-up." Such terms did little to advance the cause of boron steels. In fact, they slowed down many development efforts because fundamentals were lost sight of. Then too, the fact that for several years there did not exist adequate methods for determining the quantity of boron present in steel made many consumers suspicious of the quality of the steel and skeptical as to reproducibility of results.

It is especially worthy of note that even today, when there are available adequate and accurate chemical, spectrographic and metallographic methods of determining boron in steel, the easiest and most efficient method of determining its presence and effectiveness is the standard end-quench hardenability test.

An examination of the literature of the past sixty years will leave little doubt that boron is truly an alloying element which deserves its proper place in the field of ferrous metallurgy. Boron is not a versatile alloying element, however, exerting its effects on the mechanical properties of steel only through its potency in increasing hardenability. There are fragmentary references in the literature which tend to indicate that boron may have substantial effects on the corrosion resistance and electrical characteristics of steel, but the statements are not supported by adequate experimentation and data.

For carburized parts the proper contents of carbon and alloys other than boron must be chosen to obtain desired properties in both case and core. Since boron has little or no effect above about 0.80 per cent carbon only the carbon content can be depended upon to secure the desired hardness in the case. In the quantities used in engineering steels boron does not add to mineralogical hardness as it adds to ability to harden in depth. On the other hand, the hardenability effect of boron may make necessary an adjustment of the other elements of the steel to provide proper hardness limits at proper levels.

The effect of boron is realized only in steels which have been quenched and tempered. Therefore, if boron steels are to be used for parts which are only partially heat treated, such as integral shaft and gear designs in which only the gear is heat treated, one must check carefully the untreated portion to avoid failure in torsion because of insufficient hardness and accompanying strength.

Compare Hardenability Curves

The shape of the hardenability curve for boron steels varies somewhat from the shape of the curve for more highly alloyed steels in that it drops more abruptly from its plateau of maximum hardness. As a general rule, the boron steels match or exceed the hardness values of standard steels for the first 6 or 8-sixteenths on the end quench curve, then drop more abruptly to lower values. This characteristic makes it imperative that those who contemplate using boron steels study carefully the relationship of the hardness of the part to be manufactured and the hardenability curve of both the standard steel in use and the proposed boron steel. As long as the curves match to the equivalent section size distance on the end quench curve, there is no cause for concern regarding replacement on the basis of equivalent hardenability. It is clear, however, that the critical section size cannot be exceeded when using boron steels as it could sometimes be when using the more highly alloyed standard steels.

It has been proved by many investigators that boron in the quantity used in engineering steels does not add to mineralogical hardness of steel. Therefore, concepts held for standard steel regarding the relationship of hardness to mechanical property values still hold. In the United States most metallurgists believe that for fully quenched and tempered steels a direct relationship exists between hardness and tensile properties, between the limits of 200 and 400 Brinell. Such a relationship was adequately demonstrated during the past ten years by our experiences in the development of the lean triple alloy steels.

The surface hardness of boron steels will be the same as for standard steels of the same carbon content when fully quenched. However, since the effect of boron increases with decreasing carbon content it is desirable to keep case carbon as low as possible, consistent with the requirements of the job to be done. In the uncased hardenability curves the boron steels maintain a hardness equal to or slightly less than surface hardness to a greater depth then drop more abruptly than the curves for standard steels.

That abrupt drop means that although boron is potent in increasing the depth to which steel will harden it is not as potent in large section sizes as relatively large percentages of elements, individual or combined, such as manganese, nickel, chromium and molybdenum. Boron cannot replace those elements in others of their more important effects.

Properties are Comparable

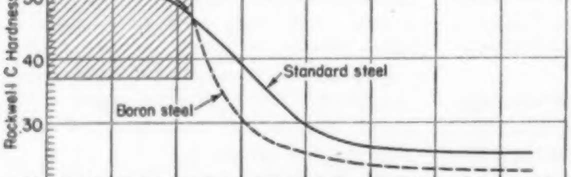
The mechanical properties of properly heat treated boron steels have been found to be comparable to like properties of conventional alloy steels when the steels have been heat treated to the same hardness values. Boron does not, however, retard softening at elevated temperatures. Therefore, boron steels must be tempered at temperatures lower than those employed for conventional alloy steels in order to secure desired hardness or strength.

The effect of boron on temper brittleness has not been exactly evaluated, but it is quite clear that where temper brittleness is an important factor boron cannot substitute for molybdenum in relieving this condition.

Users have found, too, that boron cannot be substituted for molybdenum, vanadium, or tungsten in steels which must resist creep under stress at elevated temperatures.

Since 1938 some 400 thousand tons of boron steels have been used successfully in engineering applications. In addition to reporting their adequacy from the standpoint of hardenability and ordinary mechanical properties people who have used those steels have reported the desirability of other characteristics. In the steel

(*Continued on Page* 206)



Hardenability curve comparing a boron steel and a standard steel

NEW PARTS

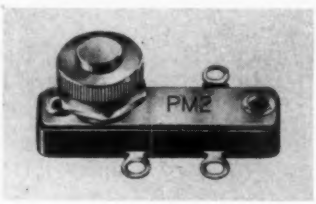
AND MATERIALS

. . . presented in quick-reference data sheet form for the convenience of the reader. For additional information on these new developments, see Page 179

MINIATURE SWITCH 1

. . . for close panel mounting

Sessions Clock Co., Tyniswitch Div., Forestville, Conn.



Compact design of this switch makes it ideal for equipment where available mounting space is limited.

Designation: PM.

Size: $1\frac{5}{8}$ in. long, $\frac{7}{16}$-in. wide, $\frac{3}{4}$-in. high; weight, 17 gm.

Service: UL rating, 15 amp at 125 v a-c, $7\frac{1}{2}$ amp at 250 v a-c; movement differential, 0.010-in. max; operating force, 7-11 oz; release force, 2-3 oz.

Design: Precision, snap-action; single-pole single-throw, either normally-open or closed, or single-pole double-throw.

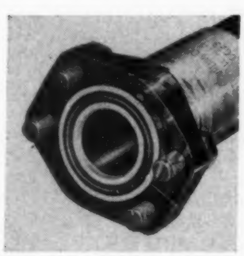
For more data circle MD 1, Page 179

SPLIT-FLANGE COUPLINGS 2

. . . for hydraulic hose assemblies

Anchor Coupling Co., Inc., 342 N. Fourth St., Libertyville, Ill.

Providing a strong bolted connection to fixed parts or a second hose, these couplings prevent leaks and eliminate threaded joints.



Size: For 1 or 2-wire braid hose from $\frac{1}{2}$ to 2 in. ID.

Service: Connecting pressure or suction lines; bolt centers are located on the centroid so that applied forces are equalized—helping to maintain a leak-proof seal; reusable; eliminate separate union connections.

Design: Split flange; straight or angle styles; bolted-clamp couplings, factory-swaged couplings, pressed-on couplings, welded tube fittings or pipe-thread adapter fittings available; O-ring seal.

For more data circle MD 2, Page 179

STEEL RINGS 3

. . . for motors, generators, sleeves, spacers

Wenthe-Davidson Engineering Co., 1459 *N.* 31*st St., Milwaukee* 8, *Wis.*

Press-lock joints on these rings eliminate flash-butt welding and maintain close tolerances on diameter and length.

Size: 4 to 12 in. diam; length up to 12 in.; ±0.005-in. tolerance on mean diam.

Service: For press-fit assemblies; reduces weight of generator yokes; manufacturing process permits complete cleanup without excessive machining.

Design: Rings are formed on specially-built hydraulic equipment; pressed lock joint.

Application: Motor housings, generator frames and yokes, lawn mower rollers, bearing spacers.

For more data circle MD 3, Page 179

LEATHER PACKINGS 4

. . . impregnated with synthetic rubber

E. F. Houghton & Co., 303 *W. Lehigh Ave., Philadelphia* 33, *Pa.*

Low friction and high strength of leather is combined with excellent sealing qualities, resilience and heat resistance of synthetic rubber.

Designation: VIM No. 1243.

Size: V packings from $\frac{1}{4}$ in. ID x $\frac{3}{4}$ in. OD to 15 in. ID x 16 in. OD; cup packings from $\frac{7}{16}$ in. cylinder diam x $\frac{1}{4}$ in. width to 12 in. diam x $1\frac{1}{4}$ in. width, with hole to suit.

Service: Sealing at temperatures to 200 F; high resiliency and pliability; resistant to oils and solvents; impregnant will not wash out of packing at rated temperature; resistant to plastic flow and abrasion; easy to assemble, due to springiness of rubber.

Design: Long-fiber mineral-tanned leather is impregnated with synthetic elastomer; bond is completed by special polymerization process.

For more data circle MD 4, Page 179

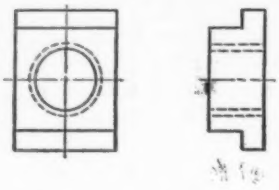
NEW PARTS AND MATERIALS

CLINCH NUT 5

. . . pierces own hole in steel sheet

Kean Mfg. Co., 23858 Kean, Dearborn, Mich.

Automatically fed to a special tooling setup on a conventional punch press, these nuts can pierce their own holes, and are then clinched to fasten rigidly in sheet metals.



Designation: Type H.

Size: For screw thread sizes 6-32 through ⅜-16 coarse-thread, and 8-36 through ⅜-24 fine-thread, Unified and American Standard.

Service: As fasteners in sheet metal; eliminate separate nut-assembly operation; adaptable to high-production operations; can be inserted in previously pierced hole, or can pierce own hole with special tooling; corners of nut body are cut diagonally and upset into sheet steel to clinch nut; for 0.037 to 0.250-in. thick sheet stock; class 2B thread fit; cadmium or zinc finish withstands minimum of 32 hr salt spray.

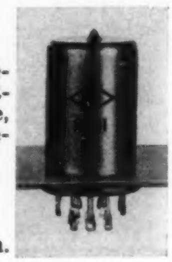
Design: Steel nut with shoulder on two ends; plain finish, or zinc or cadmium plate to min thickness of 0.0002-in.; bearing surface at right angles to hole axis within 2 deg.

For more data circle MD 5, Page 179

HERMETICALLY SEALED RELAY 7

. . . exceeds military requirements

Hart Mfg. Co., Hartford 1, Conn.



These miniature relays exceed requirements of U. S. specification MIL-R-5757 in operational shock resistance, vibration requirements, altitude requirements, and dropout voltage.

Designation: R100 series.

Size: 1.062-in. nominal diam; 2.000 in. height, base to top.

Service: Contact ratings, 2 amp at 28 v d-c, 2 amp at 115 v a-c including 400 cycles inductive, noninductive and motor loading; overload rating, 12 amp at 28 v d-c for 20 sec; shock resistance 60-64*g* with coil power consumption of 1.5 w; vibration, 40*g* with mechanical shocks to 120*g*; altitude, 80,000 ft; pull-in voltage, 18 v; drop-out voltage, 7 v; temperatures, −65 to +200 C.

Design: 4-pole, double-throw; hermetically sealed with high resistance to acids and salt spray; coil resistance, 10 to 20,000 ohm; soldered connections standard, plug-in terminals available; transit time, 1 millisec; insulation resistance, excess of 500 megohm.

Application: Aircraft, guided missiles, rockets, radar.

For more data circle MD 7, Page 179

LIMIT SWITCH 6

. . . hermetically sealed with inert atmosphere

Electro-Snap Div., Exhibit Supply Co., 4218 W. Lake St., Chicago, Ill.

Having applications where splash-proof or enclosed switches are not adequate, this switch can be used in humid or corrosive atmospheres, or in excessive dust.

Designation: H203G.

Size: $2\frac{9}{16}$ in. long, $1\frac{3}{16}$ in. wide, $1\frac{9}{16}$ in. high; weight, 0.38-lb.

Service: Limit switching with rating of 10 amp at 28 v d-c, 10 amp at 115 v a-c; hermetically sealed with inert atmosphere against humidity, corrosive atmospheres, dust and freezing; performance equal at sea level to 50,000 ft; no condensation; actuator life, 200,000 mechanical cycles, 100,000 electrical cycles, which can be increased by using lighter-action return springs; pretravel, $\frac{3}{32} \pm \frac{1}{32}$-in.; movement differential, 0.062 in., +0.025, −0.000; overtravel, $\frac{3}{32}$-in. min; operating force, 4 lb, +2, −0; difference of operating and reset points between each pole, 0.010-in. max; dielectric strength, 1000 v rms.

Design: Double-pole, double-throw, 4-circuit; single-pole, double-throw optional; tipping diaphragm action prevents ice jamming of actuator; hardened steel pivot supports diaphragm; 4 or 8-pin connector, rubber-sealed; cover is retained mechanically, sealed by induction brazing with copper.

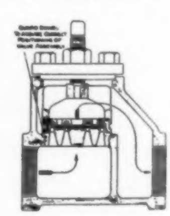
For more data circle MD 6, Page 179

CHECK VALVES 8

. . . for air-compressor discharge lines

Pennsylvania Pump and Compressor Co., Easton, Pa.

Pressure leakage through the compressor during the off cycle is prevented, pump-line pulsations are damped, and repairs to compressors can be effected without shutting down the system.



Designation: Aircheck.

Size:

Pipe Size (in.)	Length (in.)	Height (center to top, in.)	Pipe Size (in.)	Length (in.)	Height (center to top, in.)
½	6¼	5¾	2½	10¼	10¾
¾	6¼	5¾	3	10¼	10¾
1	6¼	5¾	4	12	9
1¼	7¾	7¾	5	13	9½
1½	7¾	7¾	6	16	11¾
2	8½	8			

Service: Automatic checking and line-return closure in air and gas lines to 150 psig, specials to 500 psig; temperatures from −20 to +400 F; max capacity as follows:

Valve Size (pipe, in.)	Capacity (cfm) 30 psi	Capacity (cfm) 150 psi	Valve Size (pipe, in.)	Capacity (cfm) 30 psi	Capacity (cfm) 150 psi
2½	70	130	5	465	890
3	125	240	6	770	1460
4	250	490			

Design: Two annular flat-plate stainless-steel disks are held by flat-spiral stainless springs; seat and guard locked together by retaining ring; pipe sizes ½-in. through 3-in. for threaded connections, 4 to 6 in. have flanges.

For more data circle MD 8, Page 179

JOHNSON BRONZE

SLEEVE BEARING DATA

Bearing DESIGN

JOHNSON BRONZE

SLEEVE BEARING DATA

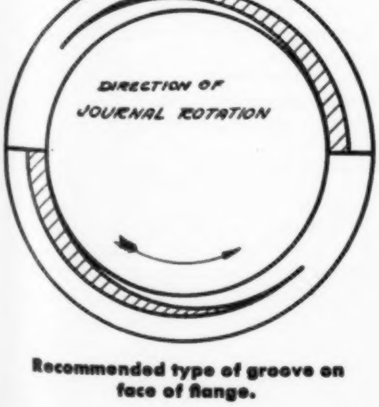
Flanged Sleeve Bearings

THE addition of a flange on a sleeve bearing often serves as a very useful device to the designer. For instance it is an ideal way of meeting the problem of end thrust. Or it can serve to locate or anchor the bearing; to act as a divider; or to form a recess to hold the lubricant. In some cases, particularly in electric motor bearings, the flange has a beveled edge to insure alignment.

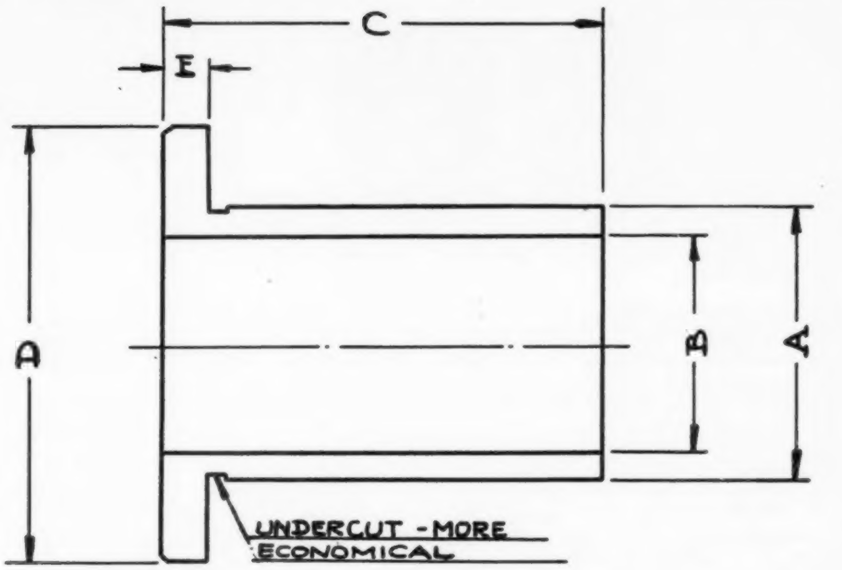
In cast bronze bearings the flange can be located anywhere on the outside diameter and can be made any thickness as required. In sheet metal bearings the location of the flange is limited to the ends and cannot be any thicker than the metal used in the body of the bearing.

In applications where the principal use of a flange is to meet a thrust condition it is well to include oil grooves on the face of the flange. In cast bronze the most popular type is the wedge shape as illustrated herewith. This permits the introduction of oil in order to produce the hydraulic pressure required to separate the flat surfaces. Sheet metal bearings usually employ a design known as "tear drop."

In applications where the operating speed is comparatively slow, it is often wise to use graphite impregnated bearings. The graphite can be included on both the inside diameter and the flange of the bearing. In some cases it is often difficult or impossible to locate the oil intake in the proper place to insure good lubrication while at the same time make it convenient and accessible. The use of flange in applications of this nature provides a reservoir for the lubricant. The oil intake can be placed in the most convenient place on the housing while the corresponding hole in the bearing can be placed where operating conditions demand.



Recommended type of groove on face of flange.



Measurements to specify when ordering Flanged Bearings.

Powder Metallurgy

When the application permits, the most economical method of producing flanged bearings is by Johnson LEDALOYL. With this method the complete bearing is formed, under pressure, to required size and tolerance. All machine work is eliminated and you gain the advantage of the self-lubricating action. Many sizes of LEDALOYL flanged bearings are standard stock items, ready for immediate delivery. Our new catalogue gives a complete listing of sizes available. Write for your copy.

How to order Flanged Bearings

When ordering or specifying flanged bearing it is important that *all* dimensions are given. This diagram can be used as a guide. When using flanged half bearings be sure to specify whether they are FULL halves or SAW CUT. Saw Cut bearings can be used where shims are included. Full cut bearings are more expensive but are necessary for precision fits. The use of an undercut under the flange lowers production costs but does not interfere with the efficient operation of the bearing.

A. Outside diameter of bearing.
B. Inside diameter of bearing.
C. Length—over-all—of bearing.
D. Outside diameter of flange.
E. Thickness of Flange.

Engineering Service

Johnson Bronze offers manufacturers of all types of equipment a complete engineering and metallurgical service. We can help you determine the exact type of bearing that will give you the greatest amount of service for the longest period of time. We can show you how to design your bearings so that they can be produced in the most economical manner. As we manufacture all types of Sleeve Bearings, we base all of our recommendations on facts free from prejudice. Why not take full advantage of this free service?

•

This bearing sheet data is but one of a series. You can get the complete set by writing to—

Johnson Bronze

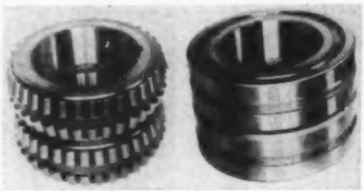
SLEEVE BEARING HEADQUARTERS
525 S. MILL ST. • NEW CASTLE, PENNA.

NEW PARTS AND MATERIALS

TAPERED ROLLER BEARINGS 9

... in large heavy-duty sizes

Tyson Bearing Corp., 1314 Oberlin Rd., Massillon, O.



Available in new size ranges, these full-roll type bearings have large capacity, rigidity and long service life.

Size: Single-row, 2 to 20 in. OD; two-row, 4 to 20 in. OD; four-row, 6 to 20 in. OD.

Service: Heavy-duty; distribution of load over full complement of tapered rolls assures smooth operation at full radial and thrust capacities; close spacing of rolls reduces cone and cup-member deflection to minimum.

Design: Full-roll type with tapered rollers; manufactured to AFBMA standards; Krupp chrome-nickel steel; high back rib is integral part of cone.

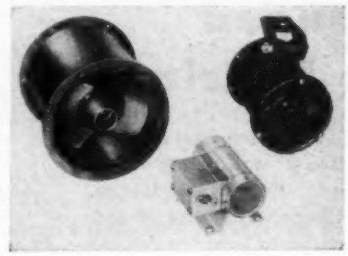
Application: Rod, bar and strip mills; pinion stands; gear drives; piercing mills; table rolls; reels; straighteners; cranes; metalworking machinery.

For more data circle MD 9, Page 179

VANE AXIAL BLOWERS 11

... for tight-fit locations

Stanat Tool & Machine Co., 47-28 37th St., Long Island City 1, N. Y.



Extensively used for cooling aircraft and electronic equipment, these blowers are engineered to meet special requirements.

Size: 2 to 8 in. diam.

Service: Supplying cooling air flow at fan speeds from 3300 to 10,000 rpm depending on pressure and volume requirements; for 400-cycle, single and three-phase a-c, 60-cycle, single-phase a-c, or 27½ v d-c current; motor is cooled by fan blast, and can be operated at top load condition.

Design: To meet requirements; motor is located in air stream; aluminum construction.

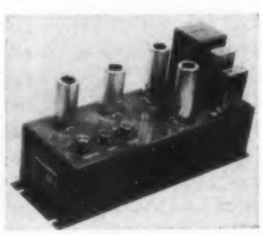
For more data circle MD 11, Page 179

SERVO AMPLIFIERS 10

... feature "breadboard circuitry"

Industrial Control Co., Straight Path and Arlington Ave., Wyandanch, L. I., N. Y.

By combining these units with pre-engineered mechanical apparatus, a servo loop can be set up in the few hours required to design and construct an interconnecting harness.



Designation: 421-A, 423-A.

Size: Chassis size, 10½ in. long, 4 in. wide, 3 in. high; ½-in. feet for mounting and fastening.

Service: For driving 400-cycle, 2-phase servo motors requiring 6 w (for 421-A) and 9w (423-A) per phase; max gain, 1000; phase adjustable through 160 deg; internal pickup, below 2 mv; damping adjustable; plate and filament power can be obtained from any power supply with no regulation, filtering or additional bias sources needed; function with any carrier frequency data system; yield servo loops of broad frequency transmission, low static and velocity errors.

Design: Set up for breadboard circuitry, with independent screwdriver controls on damping, gain and carrier phase.

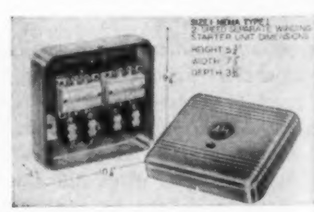
For more data circle MD 10, Page 179

MOTOR STARTERS 12

... small size and light weight

Arrow-Hart & Hegeman Electric Co., 103 Hawthorne St., Hartford 6, Conn.

"Right-angle" operation of the starter mechanism allows a smaller magnet to be used, reducing size and weight, yet multiplies the leverage and increases contact pressure.



Designation: RAR (reversing); RAS (multispeed).

Size: NEMA sizes 0, 1, 2, 3, 4.

Service: RAR—50-60 cycle, to 600 v, 1, 2 and 3-phase, 2, 3 and 4-pole; RAS—(separate winding, 2-speed) 110-550 v, 2 and 3-phase, 3 and 4-pole—(consequent pole, 2-speed, reconnected type) 110-550 v, 3-phase, 3 to 5-pole; straight-through front wiring; UL-approved; inverse-time overload protection with snap-action bimetallic thermal type relay.

Design: Right-angle operating mechanism uses bell-crank principle to move contacts horizontally; mechanical interlock prevents closing both sets of contacts; track and arc-resistant alkyd base and hood; silver alloy contacts; enclosures—NEMA type I (general-purpose), type IV (weatherproof), type VII (explosive gas and vapors), type IX (explosive dust); auxiliary interlock switches can be added.

For more data circle MD 12, Page 179

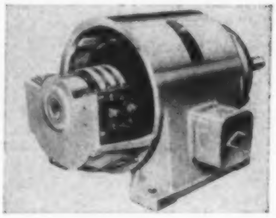

the special thermostat you need may be a Stevens standard...
now added to the standard line... waterproof, weatherproof neoprene protected Stevens thermostats
Type MH bimetal disc thermostats (left and center) for defrost control and/or cycle termination.
Positive-operating Type CH bimetal strip unit (right) for applications where moisture beads are a problem.
Now the largest line of bimetal thermostats in the industry—the Stevens line—includes Types C and M sealed in neoprene.
Featuring an electrically independent bimetal that eliminates artificial cycling and "jitters," Stevens neoprene-covered thermostats give rapid, precise control even under the most rigorous conditions. Ground shorts are eliminated . . . terminals are permanently protected against moisture or destructive contaminants. For extremes in temperatures, silicone rubber is used.
So whatever your temperature control problems, check with Stevens first—for chances are a standard Stevens thermostat will satisfy your special needs. Specify Stevens thermostats—they perform better, last longer.
A-4577
STEVENS
manufacturing company, inc.
MANSFIELD, OHIO

NEW PARTS AND MATERIALS

SLIP-RING MOTOR 13
... compact and easily mounted

Reuland Electric Co., Alhambra, Calif.

Compact arrangement of collector rings and brush assembly reduces overall dimensions to those of a standard squirrel-cage motor.



Size: $\frac{3}{4}$ to 15 hp.

Service: For applications requiring variable speed, or high starting torque on low starting current.

Design: Slip-ring type; collector rings and brushes are mounted within the motor's regular end-bell; flat end-bells permit mounting of brakes or other equipment; units can be mounted in any position; short bearing-to-bearing spacing; wound rotors are impregnated with varnish and baked to rigid mass for maximum vibration resistance; large inspection openings give easy access to collector rings and brushes.

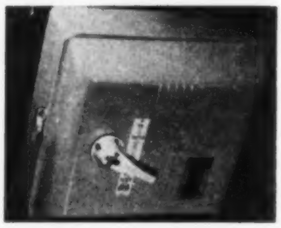
Application: Cranes, fans, printing presses, crushers.

For more data circle MD 13, Page 179

COMBINATION STARTER 15
... has components mounted "side-by-side"

Westinghouse Electric Corp., Box 2099, *Pittsburgh* 30, *Pa.*

Where mounting space is limited this horizontal design can replace the standard vertical-mount designs.



Size: NEMA sizes 0, 1, and 2.

Service: Across-the-line motor starting and overload control; handle is slamproof.

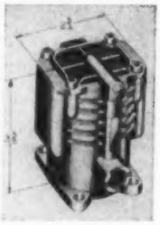
Design: Combination starter and circuit breaker with components mounted horizontally; sheet steel enclosures of NEMA types I (general purpose), IA (semi-dust-tight), and V (dust-tight); handle has separate positions for *on, tripped, off, reset* and *open cover;* special modifications, such as built-in control circuit transfer, *start-stop* pushbuttons, *hand-off-auto* selector switch, control relay, extra electrical interlocks, and inductive shunts, are also available; accommodates up to 3 separate padlocks.

For more data circle MD 15, Page 179

SOLENOID VALVE 14
... for air or oil handling

Valvair Corp., 953 *Beardsley Ave., Akron* 11, *O.*

This valve is especially recommended by the company as a pilot for its line of cylinder or diaphragm-operated valves, with operation from 150 to 300 cycles per min possible with this arrangement.



Size and Service: $\frac{1}{8}$-in. orifice for air or oil up to 125 psi; $\frac{3}{16}$-in. for up to 60 psi; connections, $\frac{1}{4}$-in. pipe tap; 110, 220 or 440 v, 60 cycles, normally open or closed; inrush current consumption, 41 w, holding, 27 w; continuous or intermittent service, with coil heat dissipation by air or fluid; operation up to 400 cycles per sec; life, excess of 10 million cycles; vertical mounting for maximum service.

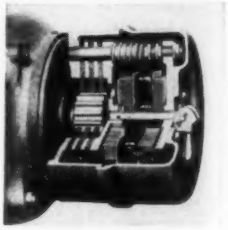
Design: 2 and 3 way, convertible by plugging exhaust port; body and cover, high-tensile zinc-alloy die casting; stainless steel seats, plunger and spring; oil-resistant Hycar seats in both plunger ends can be replaced by molding seat onto plunger; valve can be drilled through bottom for mounting on a manifold.

For more data circle MD 14, Page 179

MAGNETIC BRAKES 16
... stop motor instantly, release without drag

Dings Brakes Inc. Div., Dings Magnetic Separator Co., 4740 *W. Electric Ave., Milwaukee* 46, *Wis.*

Having no solenoids or mechanical linkages, these disk-type brakes are spring engaged and magnetically released.



Size: For NEMA type C motor flanges; styles for 203C-254C and 284C-326C.

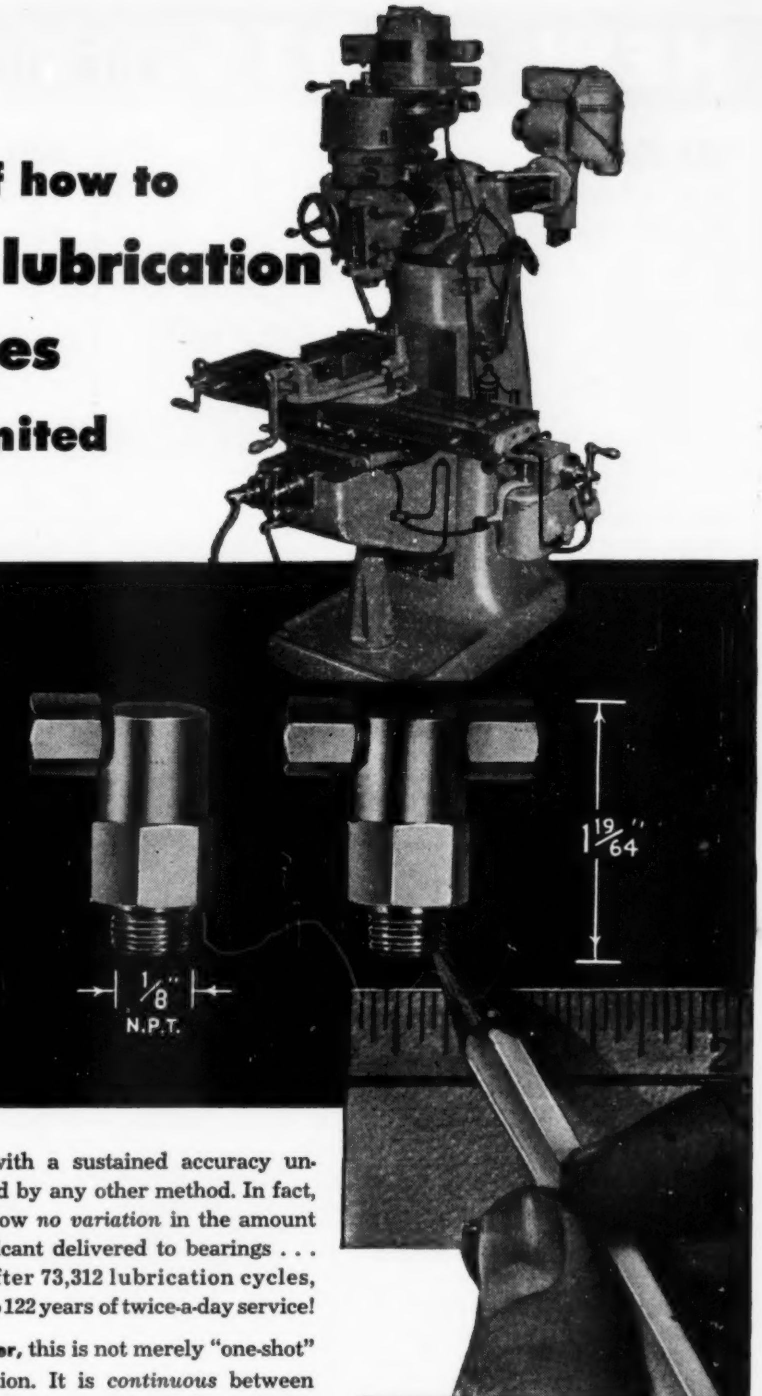
Service: Torque ratings (continuous duty) 10, 25, 50, 75 lb-ft, (intermittent) 15, 35, 75, 90 lb-ft; for 220-440 v motors, a-c or d-c; high thermal ratings; torque cannot be increased beyond ability of magnet to release; adjustable while running.

Design: Friction disk brake is spring-engaged, automatically reset when power is applied; manual release lever can disengage brake when motor is off; double set of locknuts for torque and wear adjustment; horizontal mounting, vertical if ordered; fully enclosed, dripproof, weatherproof housing or special.

Application: Machine tools, hoists, cranes, elevators, screwdowns.

For more data circle MD 16, Page 179

Solve the problem of how to design automatic lubrication into machines where space is limited



ALEMITE MIDGET Accumeter VALVES

Tiny, simple, accurate . . . Alemite Midget Accumeter Valves make it easy to design automatic lubrication into small machines, precision machines, *any* machine having many small bearings concentrated in a limited space!

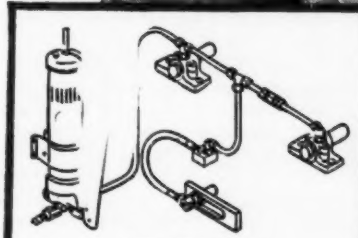
A Midget Accumeter System consists of a lubricant pump, a distribution system of tubing, and *force-feed* valves mounted on or near individual bearings. From one central point, it automatically oils all the bearings on a machine . . . *while production continues!*

Increases machine output by eliminating shutdown for lubrication. Ends the danger of errors or neglect by your customers. Makes your machines run better, cost less to maintain, last longer!

Alemite Accumeter Valves meter lubricants with a sustained accuracy unequalled by any other method. In fact, tests show *no variation* in the amount of lubricant delivered to bearings . . . even after 73,312 lubrication cycles, equal to 122 years of twice-a-day service!

Moreover, this is not merely "one-shot" lubrication. It is *continuous* between cycles, thanks to Alemite's exclusive "accumulating" feature that prolongs the discharge of lubricant to bearings!

Alemite Accumeter Automatic Lubrication is adaptable to machines of virtually every type and size. There are three different systems, to cover your full range of requirements. Write now for free bulletin giving full data. Alemite, Dept. R-12, 1850 Diversey Pkwy., Chicago 14, Illinois.



Serve Bearings of Any Type

Type O Midget Accumeter Valves can meter oil to rotary, oscillating, stationary—plain or anti-friction bearings. Up to 200 small bearings can be served by a single line system equipped with these *fixed output* valves. Require only 9/16" approximate turning radius.

ALEMITE A PRODUCT OF SW STEWART WARNER

REG. U. S. PAT. OFF.

ACCUMETER AUTOMATIC LUBRICATION
1850 Diversey Parkway, Chicago 14, Illinois

NEW PARTS AND MATERIALS

HEX LOCKNUT 17

. . . has double-locking action

Prestole Corp., 1345 Miami St., Toledo, O.

Military specification tests completed at Wright Field showed this nut to have unusually high installation, prevailing, and back-off torque, as well as excellent vibration resistance and tensile strength.

Designation: Hex Lox.

Size: Five sizes for 6-32, 8-32, 10-24, 10-32 and ¼-20 screw threads.

Service: Vibrationproof fastening; double-action spring locking with 360-deg grip; high tensile strength and torque; meets specifications of MIL-N-3337.

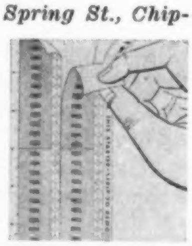
Design: One-piece nut made from 0.016 to 0.020-in. thick SAE-1060 steel.

For more data circle MD 17, Page 179

TAPE MARKERS 18

. . . identify fluid lines in aircraft

W. H. Brady Co., Dept. 273, 1602 E. Spring St., Chippewa Falls, Wis.



Conforming to military standard AND10375, these self-sticking markers are a quick and inexpensive method of identifying fluid lines.

Designation: FLU series.

Size: FLU-1 to -26 are mounted on 2¾ x 9 in. cards with 2 marker rows and 2 starter strips, marker width, 1⅛ in.; FLU-27 on 2 x 9 in. cards with 4 marker rows and 4 strips, marker width, ⅝-in.; markers are precut to ⅜, 1½, 2¼, 3, 4½ or 9 in. lengths.

Service: Designating fluid lines in aircraft; self-sticking; resist dirt, dust, oil, grease, moisture, abrasion and fungus; printed with system name, color code and symbol code according to specification AND-10375 as follows (even numbers printed with all 3, odd numbers with color code and symbol only);

—Card—	—Card—	—System—	—System—
FLU-1, -2	Rocket oxidizer	FLU-15, -16	Instrument air vacuum
-3, -4	Rocket fuel	-17, -18	Coolant
-5, -6	Fuel	-19, -20	Breathing oxygen
-7, -8	Water injection	-21, -22	Air conditioning
-9, -10	Lubrication	-23, -24	Fire protection
-11, -12	Hydraulic	-25, -26	Deicing
-13, -14	Compressed gas	-27	Warning marker

Design: Self-sticking cotton cloth tape, silicone plastic coated; bottom of card marked off as ¼-in. graduation ruler; specials furnished in any length, width and color or copy combination.

For more data circle MD 18, Page 179

HOSE CLAMP 19

. . . quickly assembled or removed

Specialty Products Co., 1965 E. 66th St., Cleveland 3, O.

Efficiently sealing hose and duct connections, this clamp is designed for convenience in handling removable equipment.

Designation: Series 21.

Size: To specifications.

Service: Clamping hose and duct assemblies; for high temperatures and pressures; corrosion resistant; high strength-to-weight ratio.

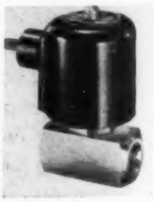
Design: Clamping band is closed and tightened by latch consisting of hex nut on threaded fastener; all stainless steel.

For more data circle MD 19, Page 179

REFRIGERANT VALVES 20

. . . smooth and quiet in operation

Marsh Instrument Co., Skokie, Ill.



Tight seating, with packless construction, these solenoid valves have brass bar-stock bodies, which prevent distortion during installation.

Designation: Marsh-Electrimatic, types 60, 65, 67.

Size: Orifice size, type 60—3/16-in., type 65—⅜-in., type 67—½-in.; electrical connections for ½-in. conduit; 24-in. leads;

—Model—		Inlet and Outlet (in.)	Body Length (in.)	Height (in.)
60	65	⅜ FPT	2½	4¼
60-1	65-1	⅜ sweat	4	4¼
60-2	65-2	½ sweat	4	4¼
	67	½ FPT	2⅝	4⅞

Service: Max pressures, type 60, 150 psi, type 65 and 67, 200 psi; coils are moisture and frostproof; nonmagnetic valve stem and seat; tight seating; UL-approved; coils available for 20 to 440 v a-c, 6 to 230 v d-c; capacity in tons of liquid Freon-12 as follows;

Pressure drop (psi)	2	3	4	5	6
Type 60	3.10	3.79	4.36	4.91	5.35
Type 65	8.3	10.2	11.5	13.6	14.4
Type 67	12.0	14.6	17.0	19.2	21.0

Design: Normally closed, open when coil is energized; type 60 is direct-acting, 65 and 67, pilot-operated; solid brass bar-stock bodies; impregnated coils.

For more data circle MD 20, Page 179

ADLAKE RELAYS AT WORK—One of a series of advertisements on specific ADLAKE applications.

for ASSURED SAFETY

Baker Perkins uses ADLAKE mercury relays

Because of their dependability and positive action, ADLAKE Relays are used in the burner control circuits of Baker Perkins bakery equipment, where they have an important "watchdog" job to do (see panel).

It's no wonder that Baker Perkins, like so many other leading manufacturers, specifies ADLAKE Relays. For they are designed and built to meet the most exacting requirements. Their mercury-to-mercury contact prevents burning, pitting and sticking, and their sturdy construction armors them against outside vibration or impact. And most important of all, *they require no maintenance*, for they are hermetically sealed.

For the full story on the part ADLAKE Relays can play in *your* business, just drop a card to The Adams & Westlake Company, 1128 N. Michigan, Elkhart, Indiana. No obligation, of course.

EVERY ADLAKE RELAY BRINGS YOU THESE ADVANTAGES:
HERMETICALLY SEALED—dust, dirt, moisture, oxidation and temperature changes can't interfere with operation.
MERCURY-TO-MERCURY CONTACT—prevents burning, pitting and sticking.
SILENT AND CHATTERLESS • REQUIRES NO MAINTENANCE
ABSOLUTELY SAFE

Adlake
TRADE MARK

THE Adams & Westlake COMPANY
Established 1857 • ELKHART, INDIANA • New York • Chicago
Manufacturers of ADLAKE Hermetically Sealed Mercury Relays

NEW PARTS AND MATERIALS

GLASS CLOTH INSULATION 21

. . . impregnated with silicone rubber

General Electric Co., Chemical Div., Coshocton, O.

Presently used for motor and cable insulation, this type of cloth has possibilities in many applications requiring good electric insulating qualities with high heat resistance.

Form: Continuous sheet of glass cloth impregnated with silicone rubber.

Size: 36 in. wide; 4 to 10 mils thick.

Service: Electric insulation with high heat resistance; good flexibility at high temperatures.

Properties: Dielectric strength (v per mil, ¼-in. electrodes, short time) 950 initially, (28-day tests) 605 at 200 C, 619 at 240 C, 600 at 275 C; power factor (60-cycle), 0.022 at 25 C, 0.025 at 100 C; dielectric constant, 2.5 at 25 C, 2.8 at 100 C.

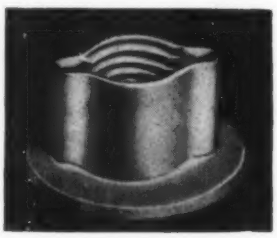
For more data circle MD 21, Page 179

LOCKNUT 22

. . . holds by concentrated spring action

E. A. Bessom Corp., 127 *Margin St., Salem, Mass.*

Longitudinal ribs on the barrel cause a concentrated spring action which locks the nut under vibration conditions.



Designation: Con-Torque, round base.

Size: Also available in 2-56, 2-64, 5/16-18, 5/16-24, ⅜-16, and ⅜-24 sizes, standard sizes are as follows:

Screw Size (NC)	Screw Size (NF)	Height (in.)	Base Diam (in.)
4-40	4-48	5/32	5/16
6-32	6-40	3/16	5/16
8-32	8-36	7/32	⅜
10-24	10-32	17/64	7/16
¼-20	¼-28	11/32	9/16

Service: Locking under vibration, moisture conditions and temperature extremes; locks in any position of screw; retains grip after many removals; class 3 fit, 70% or better thread; thread contact is over full arc.

Design: One-piece steel or bronze flange and dieset-ribbed barrel, heat-treated to spring temper; zinc plated, or special finish; 180-deg and 90-deg base plate styles also available.

For more data circle MD 22, Page 179

RUBBER COMPOUND 23

. . . restrains oil shrinkage and swelling

Acushnet Process Co., New Bedford, Mass.

Developed to satisfy critical operation and life conditions, this compound, having minimum swell and shrinkage, is produced under rigid quality-control procedures.

Designation: X 1692.

Form: Buna N rubber compound as packings, bushings, seals or couplings.

Size: To specifications.

Service: Sealing or coupling applications; minimum swell in low aniline point oil; minimum shrinkage in high aniline point oil; will not corrode or damage aluminum, magnesium and other alloys, or stainless steel; compression set (70 hr at 250 F)—38% (original deflection), 12% (original thickness); neither tacky nor decomposed after aging in oils with aniline points of 157.1 F and 255 F for 70 hr at 300 F.

Properties: Hardness (Shore A durometer), 70; tensile strength, 1400 psi; elongation, 400%.

For more data circle MD 23, Page 179

FLOAT VALVES 24

. . . provide large capacity in compact sizes

McDonnell & Miller Inc., 3500 *N. Spaulding Ave., Chicago* 18, *Ill.*

Valve opens to full capacity with small float drop, and provides positive sealing.

Designation: No. 18, 118, 518.

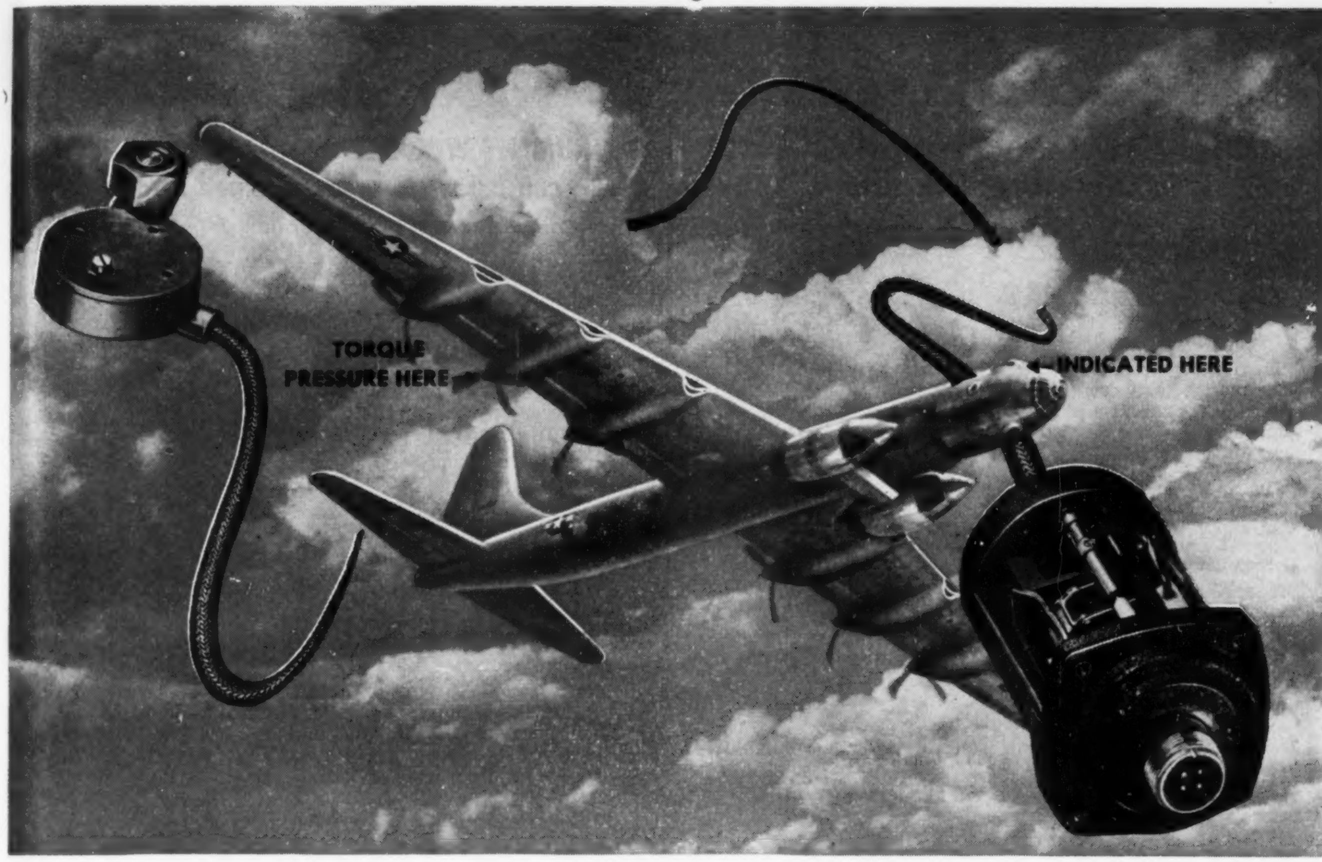
Size: No. 18, 8 13/16 in. long, 2 17/32 in diam; No. 118, 7 9/16 in. long, 2 17/32 in. wide, 4 33/64 in. high; No. 518, 9 7/32 in. long, 3 31/32 in. wide, 4⅜ in. high; all have ⅝-18 thread and nut for mounting and ⅛-in. pipe tap for ⅛-in. pipe or ¼-in. OD tubing.

Service: For line pressures to 100 psi; approximate capacity with ⅛ in. pipe supply line as follows:

Water Pressure (psi)	Discharge (lb per hr)	Water Pressure (psi)	Discharge (lb per hr)
10	500	60	1200
20	700	80	1450
40	950	100	1500

Design: No. 18 consists of valve mechanism and float in line; No. 118 has float mounted at right angles to valve so valve proper is above water level, and has supplementary diverter plate to direct incoming water downward; No. 518 has float and valve in die-cast chamber with cover, and supply connection at side for external applications; copper or brass parts; neoprene valve seat; disassembles by removing single brass pin.

For more data circle MD 24, Page 179



A NEW APPROACH IN AIRCRAFT TORQUE PRESSURE INSTRUMENTATION

Provides complete freedom from the attacks of sand, dust, fungus, salt fog and humidity

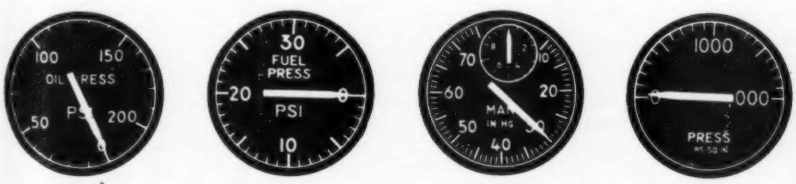
LOOK AT THESE EXCITING "FIRSTS" . . .

- Hermetically sealed and helium protected pressure and electrical elements.
- Nispan "C" pressure elements eliminating bimetallic compensation.
- Higher precision in remote transmission.
- Smaller and lighter in weight with tear-strip ease of disassembly.
- External adjustment through hermetic seal.
- Isolating link and pressure pick up for temperature extremes of —85 and +230 F.

In this Synchro Torque Pressure Transmitter one element measures the pressure and the other the ambient outside pressure. Together, these two Nispan elements measure the applied pressure in a sealed case charged with an inert gas, and transmit an electrical signal through a glass-to-metal connector.

This refreshing solution to a difficult problem leads a whole flight of new direct-connected clamp-on pressure indicators for oil, fuel, manifold, and hydraulic pressures.

United States Gauge, Division of
American Machine and Metals, Inc.,
Sellersville, Pa.



USG Oil Pressure Gauge — USG Fuel Pressure Gauge — USG Manifold Pressure Gauge — USG Hydraulic Pressure Gauge

USG — UNITED STATES GAUGE

Quality Gauges Engineered for Enduring Accuracy

PRODUCTS OF UNITED STATES GAUGE . . . Absolute Pressure Gauges • Aircraft Instruments • Air Volume Controls • Altitude Gauges • Boiler Gauges Chemical Gauges • Mercury, Gas, and Vapor Dial Thermometers • Glass Tube and Industrial Thermometers • Flow Meters • Inspectors' Test Gauges Precision Laboratory Test Gauges • Marine, Ship and Air-Brake Gauges • Voltmeters • Ammeters • Welding Gauges

OTHER DIVISIONS OF AMERICAN MACHINE AND METALS, INC., AT SELLERSVILLE, PA.: GOTHAM INSTRUMENTS AND AUTOBAR SYSTEMS

NEW PARTS AND MATERIALS

FILTERS 25

. . . in large-capacity stock sizes

Marvel Engineering Co., 625 *W. Jackson Blvd., Chicago* 6, *Ill.*

Two new sizes, of 75 and 100 gpm capacity, are believed to be the highest capacity stock filters now made.

Designation: Synclinal.

Size: 75 gpm model, 12½ in. long, 6 in. diam, coupling nut for 2½ in. pipe; 100 gpm model, 14 in. long, 6½ in. diam; mesh sizes, 30 to 200 in all models.

Service: Filtration of noncorrosive liquids; eight models in total line, for 5, 8, 10, 20, 30, 50, 75 and 100 gpm capacity; can be disassembled, cleaned and reassembled at working location.

Design: Line or sump type; corrugated wire-mesh filter element.

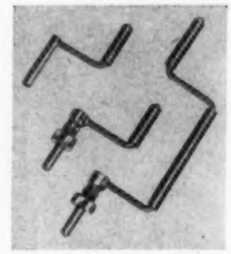
Application: Hydraulic equipment; low-pressure liquid recirculating machinery.

For more data circle MD 25, Page 179

INSTRUMENT WINDERS 27

. . . can be set to predetermined torque

J. A. Campbell Co., 645 *E. Wardlow Rd., Long Beach* 7, *Calif.*



Discomfort of twisting a key to wind the instrument is eliminated. Two of three models have torque-wrench arrangement to prevent overwinding.

Designation: Winder-Upper, types A, B, C.

Size: Type A, 3½ in. long, 2¼ in. wide; type B, 3⅞ in. long, 2¾ in. wide; type C, 8¾ in. long, 2¾ in. wide.

Service: For square key shafts; torque-wrench models prevent overwinding.

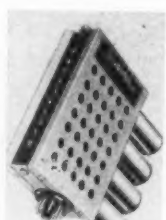
Design: Type A, simple crank; type B, crank with torque chuck having slip-grip arrangement; type C, with brace handle torque chuck to prevent wobble; metal parts are triple chrome plated; red Catalin plastic handles; hardened shaft ends.

For more data circle MD 27, Page 179

ELECTRONIC COUNTERS 26

. . . operate at near ultrasonic rate

Stratex Instrument Co., 1861 *Hilhurst Ave., Los Angeles* 27, *Calif.*



These scaler counters and totalizers record impulses at rates as high as 100,000 per sec.

Designation: Detectron, models TU-9, TU-100.

Size: 5½ in. long, 5 in. wide, 1½ in. thick; weight, 10 oz.

Service: TU-9 records impulses to 30,000 per sec, TU-100 to 100,000 per sec; direct reading from internally illuminated panel; instant external resetting; input pulse, 100 v negative (same for output pulse); units can be cascaded interchangeably (TU-100 used as first decimal counting unit) to make up a complete register, or mechanical registers can be inserted after fourth scaling unit (where count is scaled down to 600 per min); power supply, 200 to 400 v a-c at 12 ma; resolving time, 3 microsec per pulse pair; can be placed in circuit with sylphon bellows, precision switches, crystal oscillators, Geiger-Mueller tubes, strain gages.

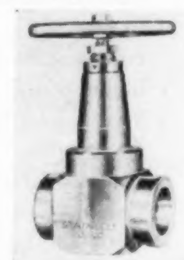
Design: Four No. 5963 computer tubes and associated circuit shielded by aluminum-alloy case; standard octal base plug.

For more data circle MD 26, Page 179

GLOBE VALVES 28

. . . of forged stainless steel

A. F. Hoppe Engineering Inc., Greensboro, Ind.



Now being used on the first atomic-powered steam turbine, these valves are welded closed for use with poisonous liquids and gases.

Size: ½ to 6 in. pipe size; typical dimensions—4-in. size, 12-in. end to end, 19 in. center to top—1 in. size, 4½ in. end to end, 5½ in. center to top.

Service: For high-pressure fluid lines, poisonous fluids and gases; rated at 10,000 psi; corrosion resistant.

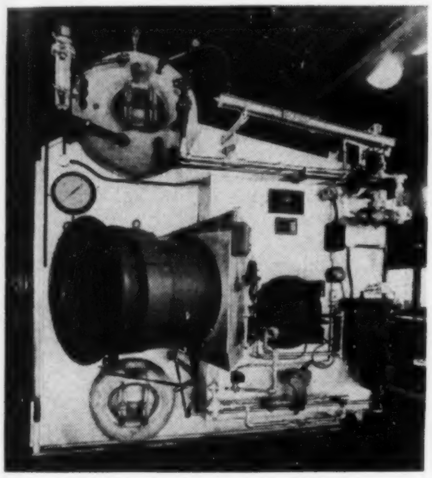
Design: Horizontal or angle styles; parts are machined and welded closed; packing-gland bolts are eliminated; polished internally to help prevent corrosion; furnished with bolted or welded caps; screwed, flanged or socket-weld ends; body of stainless steel, types 302, 304, 321 or 347; stems and disk are 322 stainless.

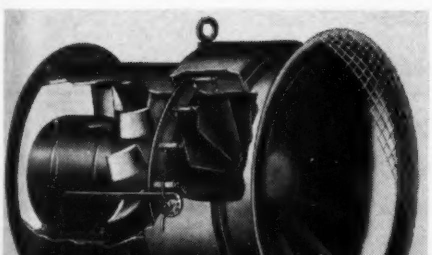
For more data circle MD 28, Page 179

ADVERTISEMENT

TOTALLY ENCLOSED MOTOR COOLED BY AXIAL AIR FLOW

In the type EMD Forced Draft Blowers built by L. J. Wing Manufacturing Company, Linden, New Jersey, a highly efficient means for cooling the driving motor is provided. Since these Wing blowers are so extensively used by leading manufacturers of equipment designed for combustion applications, totally-enclosed dust-tight motors are required. Axial flow fan wheels mounted directly on the motor shafts provide air-over-motor cooling.





Typical installation of an L. J. Wing blower on a Foster Wheeler unit. (Bottom) Cutaway view of blower shows totally-enclosed motor and the axial flow airfoil fan wheel for air-over-motor cooling.

The motor used is a 5hp, 60 cycle, 3 phase, 220/440 volt, 1740 rpm unit of the squirrel-cage type, designed by Star-Kimble, which has been supplying motors to L. J. Wing for over 25 years. Prelubricated bearings and extra insulation—standard features of Star-Kimble squirrel-cage motors—are of special importance in long motor life with little maintenance attention in hot, dusty locations.

Another advantage, resulting from recent redesign of Star - Kimble squirrel-cage motors, is the elimination of crevices that catch dust and dirt.

Information on sizes and types of these motors is contained in Bulletin B-201, available on request from Star-Kimble Division of Miehle Printing Press and Manufacturing Company, 201 Bloomfield Avenue, Bloomfield, New Jersey.

YOU GET THESE 5 ADVANTAGES

when you use Star-Kimble Squirrel-Cage Motors

1. *Sturdy construction.* Frames through NEMA 364 are of rigid cast construction, with more internal ribs than are used in conventional design, giving added strength and better heat dissipation. Larger frames are made of steel.
2. *Simplified maintenance.* Double-width cartridge bearings are prelubricated—need no attention for many years.
3. *Versatility of mounting.* Standard NEMA horizontal, D Flange, C Face, P Ring. Motors through frame 320 can be made dripproof in any mounting position—simply by rotating end shields.
4. *Extra insulation.* Safety factor is provided for possible rises above rated temperature—motor life is lengthened, maintenance reduced.
5. *Simplified design.* Lines are smooth, with no dust- and dirt-catching crevices. Fewer basic parts—greater interchangeability. Basic parts stocked for immediate shipment.

Star-Kimble Squirrel-Cage Motors are available in NEMA frames 203-505, and in NEMA Designs "B", "C" and "D." Also in larger sizes. Write for Bulletin B-201.

*Demanded by industry for tough start-and-stop jobs

STAR-KIMBLE

MOTOR DIVISION OF

MIEHLE PRINTING PRESS AND MFG. CO.

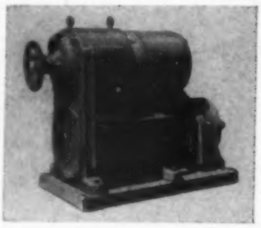
201 Bloomfield Avenue Bloomfield, New Jersey

NEW PARTS AND MATERIALS

VARIABLE-SPEED DRIVE 29

. . . *with integral motor and reducer*

Reeves Pulley Co., Columbus, Ind.



This self-contained power drive saves space, since all units are contained in one package.

Designation: Model 8000.

Size: No. 8342, 47⅜ in. long, 30¼ in. wide, 43½ in. wide; No. 8402, 50¼ in. long, other dimensions same; NEMA motor frame sizes 364, 365, 404 or 405.

Service: Variable speed at 25 hp; 2:1 to 6:1 speed range.

Design: Combination motor, speed varying mechanism, and helical gear reducer if required for low output speeds; choice of standard constant-speed motor; horizontal or vertical; handwheel control standard, electric or completely automatic hydraulic controls special; tachometer can be attached.

For more data circle MD 29, Page 179

ROTARY VANE PUMPS 30

. . . *with internal or external bearings*

Worthington Pump & Machinery Corp., Harrison, N. J.

Pressure of the liquid being pumped maintains contact of the vanes against the liner.

Size and Service: Pumping nonlubricating or abrasive liquids with external bearing model; heavy-duty pumping of clean liquids with internal bearing model; max operating pressure, 150-300 psi; max viscosity, 20,000 SSU;

Size	Pipe Openings Suction (in.)	Pipe Openings Discharge (in.)	Ratings (nominal) Capacity (gpm)	Ratings (nominal) Speed (rpm)
1	2	2	25	850
2	2	2	50	600
3	3	3	100	475
4	4	4	200	425
5	6	6	300	350
6	8	6	500	300
8	10	8	750	250
10	12	10	1000	210

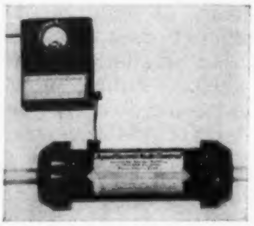
Design: Positive-displacement, sliding-vane type; 6 vanes, self-compensating for wear; removable liners; end-plates and vanes can be reversed for wear; standard or bronze fitted, or all-bronze construction; model VE has external bearings, VR has internal bearings, VRH has internal bearings with suction and discharge openings in same plane; VEU, VRU and VRHU are same with integral relief valves.

For more data circle MD 30, Page 179

ELECTRIC WATER CONDITIONER 31

. . . *prevents formation of hard boiler scale*

Aqua Electric Scale Control Inc., 2028 *E.* 22*nd St., Cleveland* 15, *O.*



Eliminating chemicals and boiler compounds, these units consume per month about 15 cents worth of electricity.

Size: For ½, ¾, 1, 2, 3 or 4-in. pipe.

Service: Prevention of hard scale formation in boilers and heat exchangers; water is passed through electric field which charges colloidal particles and dissolved salts; charged particles are no longer attracted to, and do not build up on tubes, castings and metal surfaces; current 110 v, 60 cycles a-c.

Design: Transformer in control unit sets up electric field within plastic cylinder spliced into incoming water line.

Application: Boilers, heat exchangers, air-conditioning units, bottle washers.

For more data circle MD 31, Page 179

MOTOR CONTROL UNITS 32

. . . *raintight for outdoor service*

Westinghouse Electric Corp., Box 2099, *Pittsburgh* 30, *Pa.*

Starter and circuit breaker are mounted on removable panels for easy servicing.

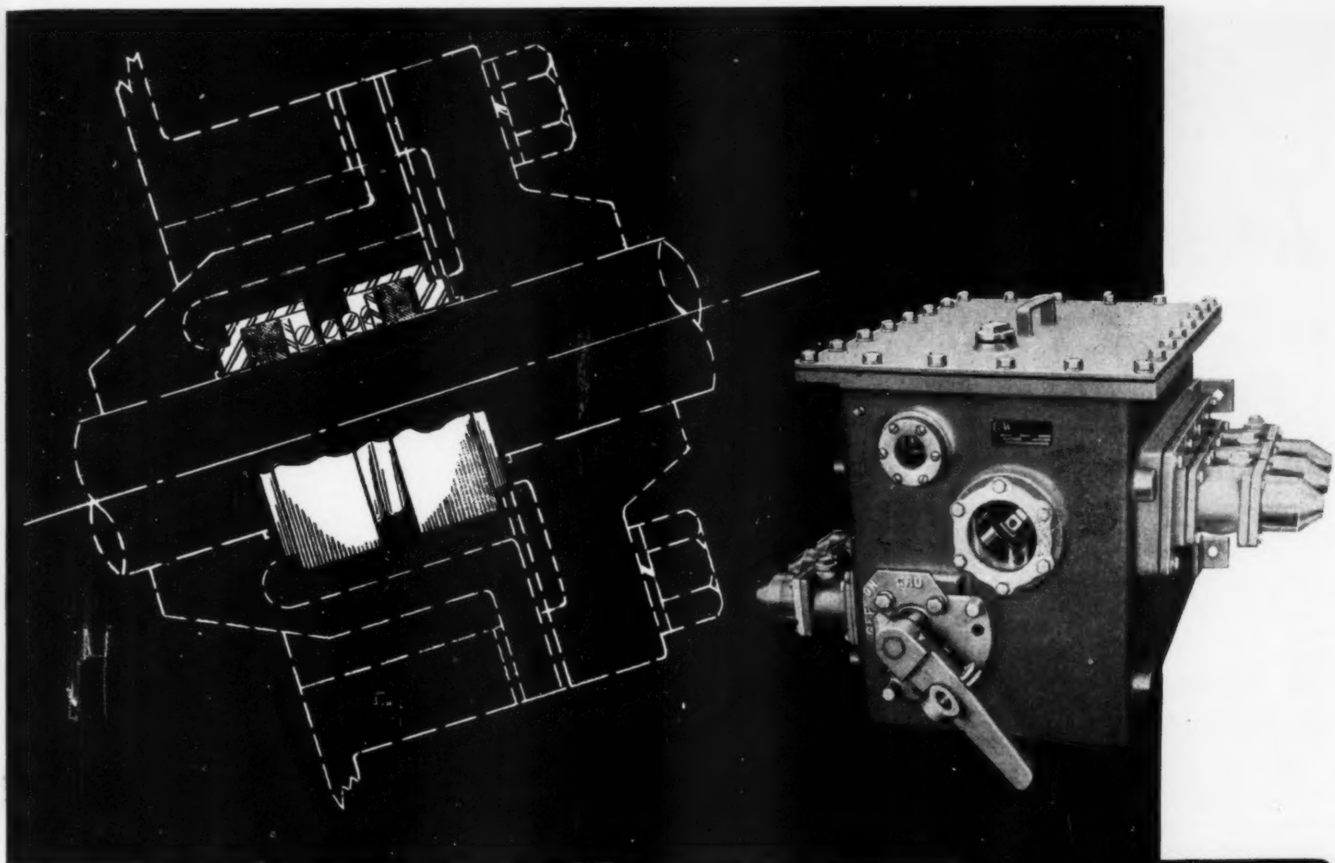
Designation: 11-206-NR.

Size: NEMA sizes 1 through 5.

Service: Motor starting and overload control in outdoor conditions; sealed enclosure is raintight; 220 or 440 v, 60 cycles a-c, with special coils for up to 600 v; enclosure is corrosion resistant.

Design: Across-the-line starter and circuit breaker are mounted on a removable steel panel; sheet steel enclosure, Bonderized and finished in gray enamel; Neoprene gasket; protective rain shield at top of enclosure; slamproof handle for circuit breaker, with *on, tripped, off, reset* and *open cover* positions; raintight *reset* button; raintight *start-stop* pushbutton or *hand-off-auto* selector switch; 1 or 2 control relays, a third overload relay, or a control transformer can be added.

For more data circle MD 32, Page 179



Shaft-Sealing Certainty

for all G & W switches—with

ROTARY SEALS

Even after complete submersion for protracted periods, or when exposed to weather extremes, the Load Break Oil Switches made by G&W Electric Specialty Co., Chicago, must be in perfect condition to make an alternate source of power immediately available the instant cable trouble occurs. Any Seal failure at the point where the operating shaft extends through the wall of the hermetically sealed tank might be serious.

To assure the *Shaft-Sealing Certainty* they must have, G&W uses ROTARY SEALS on their entire line of Switches . . and, quoting from their bulletin, "ROTARY Oil Seal Rings have been used on Type RA Switches since 1930 with no reports of any leaks."

The experience of our engineers in adapting the ROTARY SEAL principle to the particular requirements of manufacturers in many fields is at your service to determine the best method of assuring *Shaft-Sealing Certainty* for your application. It is usually best to call us in at the drawing-board stage—we can often point out simpler and more practical ways to approach the Shaft-Sealing problem.

THE ROTARY SEAL PRINCIPLE

is the original approach to a practical solution of a universally troublesome problem. Our booklet "SEALING WITH CERTAINTY" explains and illustrates the principle. We're glad to send it to you without obligation.

THE ROTARY SEAL COMPANY

mechanical seals for rotating shafts

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CHICAGO 14, ILLINOIS, U. S. A.

MEN OF MACHINES

Robert K. Norton

With a background of nearly twenty years of experience in engineering design and factory management, **Robert K. Norton** has been promoted to the position of assistant engineer in charge of letterpress development for Harris-Seybold Co., Cleveland. Mr. Norton attended the Henry Ford Trade School and Wayne University. He was affiliated with the Ford Motor Co. and Pipe Machinery Co. before joining Harris-Seybold in 1946. Since that time he has done intensive research on letterpress equipment, having helped develop the Harris Model TRG, a two-color rotary typographic press now widely used for fine publication and commercial printing. More recently Mr. Norton assisted in the development of the two-color rotary TRG-C press, which is for high-speed carton printing exclusively. Mr. Norton is a member of the Cleveland Engineering Society.

◆

John L. Fuller

Reliance Electric & Engineering Co., Cleveland, has promoted **John L. Fuller** to the newly created administrative post of manager of research and technical services. Following graduation from Case Institute of Technology with a B. S. degree in electrical engineering, Mr. Fuller joined Reliance as an engineering trainee in July, 1936. He was a junior engineer for the next five years and during this period contributed to the development of the V-S drive and the broadening of its in-

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Precision-fabricated parts can have varying degrees of hardness, tensile strength, and elongation to meet individual, S.A.E. or A.S.T.M. specifications.

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dustrial applications. In 1941 he was advanced to the position of experimental engineer, supervising all experimental work, including the design and construction of enlarged laboratory quarters and testing facilities. In 1947 he was appointed manager of large motors engineering and held that position until September, 1950, when he was named technical co-ordinator in the engineering department. Mr. Fuller is a member of the American Institute of Electrical Engineers and the Cleveland Engineering Society.

♦

The newest addition to MACHINE DESIGN's editorial staff is Assistant Editor **Leo F. Spector,** whose background is discussed in "Over the Board," Page 4.

Leo F. Spector

♦

Joseph H. Lancor Jr. has been appointed director of the transducer division of Consolidated Engineering Corp., Pasadena, Calif. He will supervise and direct the design and development of transducers of various types including vibration pickup, accelerometers and pressure pickup. Since 1947 Mr. Lancor has been director of product engineering of Vitro Corp. of America and before that served as a research engineer for Sperry Gyroscope Co.

♦

Westinghouse Electric Corp., Pittsburgh, has appointed **R. C. Bergvall** as manager of engineering, defense products. He previously held a similar post in the industrial products division,

♦

A. F. Gagne Jr., who has contributed a number of articles to MACHINE DESIGN, recently joined Modern Design and Engineering Co., Vestal, N. Y., as special projects engineer. He comes to this company from E. I. du Pont de Nemours & Co. Inc., where for ten years he served as machine development engineer.

♦

A. J. Mallinckrodt was appointed chief engineer of United States Air Conditioning Corp., Minneapolis. He was formerly manager of engineering for Baker Refrigeration Corp.

♦

The appointments of **R. A. Millermaster** as manager of the development department and **C. W. Kuhn** as director of development engineering has been announced by Cutler-Hammer Inc., Milwaukee. Mr. Millermaster joined the company in 1927 and for the

past five years has been assistant manager of the development department. Mr. Kuhn came to the company in 1923 as a student engineer, became a member of the development department two years later, and since 1946 has been assistant manager of that department.

♦

Returning to the firm which he first joined as a draftsman in 1940, **Harvey J. Hincker** has been named chief engineer of the Automatic department of the Automatic Transportation Co., Chicago. Most recently Mr. Hincker was associated with the Buda Co., Harvey, Ill., as supervisor of design and research in the materials handling division. Concurrently, announcement was made by Automatic of the promotion of **Edward W. Gammell** to assistant chief engineer of the Transporter department and **Jerome W. Tiskus** to assistant to Mr. Hincker.

♦

Frederick Schultz was recently named chief engineer for all products manufactured in the Ilion, N. Y. plants of Remington Rand Inc.

♦

Dr. Chung Hua Wu has been appointed as a full professor at the Polytechnic Institute of Brooklyn. Since 1947 he has been associated with the Lewis Flight Propulsion laboratory of the National Advisory Committee for Aeronautics at Cleveland, carrying out fundamental research on jet engines. At Polytechnic, Dr. Wu will direct the first research project on turbo machines and jet engines undertaken by the Department of Mechanical Engineering.

♦

Consolidated Vultee Aircraft Corp., San Diego, Calif., recently named **Quentin G. Turner** manager of industrial engineering, Convair Guided Missile division, Pomona, Calif.

♦

Vonnegut Moulder Corp., Indianapolis, has appointed **Oliver S. DeHaven** to reassume the duties of chief engineer. He served in this capacity from 1941 to 1948, and directed the research and experimental work on Vonnegut products.

♦

With nearly 20 years of experience in the gas turbine and aircraft engine field, **Eugene S. Clark** has joined the Lycoming-Spencer division of AVCO Manufacturing Corp., Williamsport, Pa. as project engineer on gas turbine developments.

♦

Maurice Harp has joined the engineering staff of Lenkurt Electric Co., San Carlos, Calif., manufacturer of carrier equipment. He will serve as applications engineer for the development of f-m and single-sideband space-carrier equipment.

♦

Associated with the company since 1929 and having served as director of plant engineering for the last six years, **William Irrgang** was recently elected executive vice president of Lincoln Electric Co., Cleveland.

THE ENGINEER'S Library

Heat and Thermodynamics

Third edition by Mark W. Zemansky, professor, City College of New York; published by McGraw-Hill Book Co., New York; 465 pages, 6 by 9 inches; clothbound; available from MACHINE DESIGN, *$6.00 postpaid.*

Requiring only a knowledge of calculus and general physics, this book is written at an intermediate level to provide an introduction to the subject. The first ten chapters treat fundamentals, including the concepts of temperature, heat, internal energy and entropy, and the first and second laws of thermodynamics. The remaining nine chapters deal with physical, chemical and engineering applications in sufficient detail to show how the general principles are applied to specific cases. Besides several revisions and regroupings of subject matter to bring the book up to date and include new material, the chapter on heat transfer has been expanded to include streamline flow, Poiseuille's law, Reynolds number and turbulence, natural and forced convection, and the theory of the single-current heat exchanger.

□ □ □

Production Methods

The Grinding Wheel—A Textbook of Modern Grinding Practice: *By Kenneth B. Lewis; published by the Grinding Wheel Institute, Greendale, Mass.; 409 pages, 6 by 9 inches, Leatherette bound, semihard covers; price, $3.50.*

A Treatise on Milling and Milling Machines: *Third edition compiled by Mario Martellotti, development engineer; published by the Cincinnati Milling Machine Co., Cincinnati 9, Ohio; 910 pages, 6 by 9 inches, clothbound; price, $8.00 postpaid.*

Compiled as comprehensive reference sources in their respective fields, both these books fully describe processes, machines and tools. Generally speaking, both books emphasize production considerations, and the information contained is mainly of value to the production engineer. Nevertheless, there is much material of primary importance to the designer concerned with proper design of parts for either type of operation.

The Grinding Wheel is in the nature of an encyclopedia, containing boiled-down and condensed, rather than exceptionally detailed, information. Having a total of 28 chapters, the book first considers fundamentals; principles of grinding; grinding wheel com-

positions, shapes and sizes; grinding fluids; and evaluation of surface quality. A number of chapters discuss various grinding machines and grinding methods, such as cylindrical, centerless, surface, and internal grinding, cutting-off, thread and gear grinding. Proper design of parts for grinding and production costs are discussed in two chapters.

A Treatise on Milling is a much more detailed and extensive work, although the subject matter is handled in a similar manner. After several chapters on milling machines, accessories and milling cutters, basic milling theory is reviewed, together with a discussion of chip formation, surface finish and cutting fluids. Use of the milling machine in producing different shapes and parts (such as helical surfaces, cams, dies and molds) is considered; use in toolroom work and production milling is also presented in two chapters. Fixtures and fixture design, and estimating costs are subjects of the two final chapters.

Association Publications

Recommended Practice for the Welding of Steel Castings: Based on studies carried out by the Steel Founders' Society, member companies and professional research engineers, this 40-page report establishes quality standards and welding procedures for cast-weld construction techniques (welding steel castings together to form an integral unit), and for composite fabrication (fabrication of structures through the welding of steel castings and wrought steel products). The book, in three sections, covers welding methods, types of electrodes, and recommended welding procedure for carbon-steel and low-alloy steel castings. Steel Founders' Society of America, 920 Midland Bldg., Cleveland 15, O., is the publisher of this 8½ by 11 inch booklet. Cost is 35 cents each.

A Survey of Creep in Metals—NACA Technical Note 2516: Numerous theories about creep phenomena (deformation of metals under stress over a period of time) and extent of current knowledge on the subject are summarized. Various possible mechanisms by which creep occurs, both in single crystals and polycrystals, are described. Especially worth noting is the extensive bibliography of 199 references.

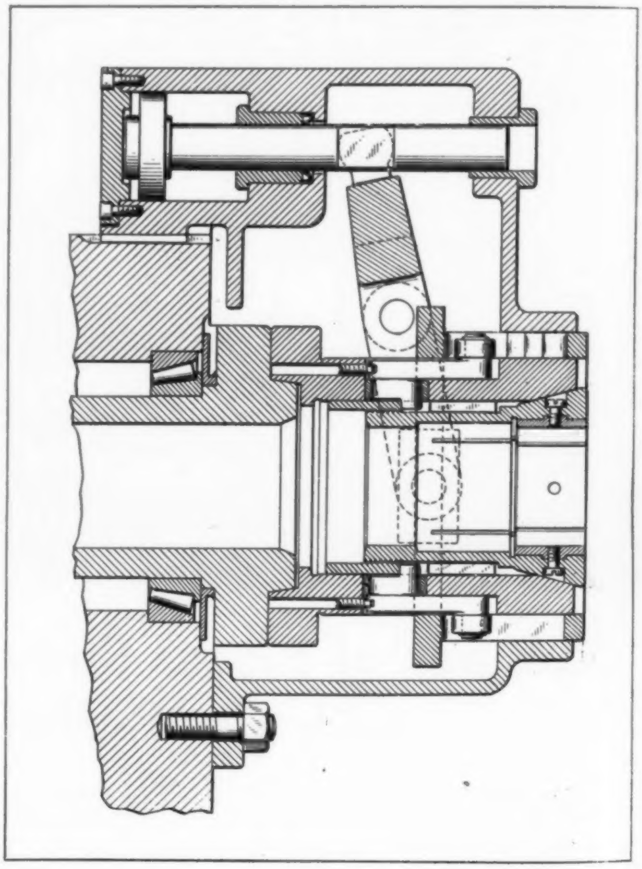
The report is 66 pages long, 7¾ by 10¼ inches in size, with paper covers and side-stapled. Copies are available from National Advisory Committee for Aeronautics, 1924 F St., Washington 25, D. C.

Experimental Investigation of Oil Film Pressure Distribution for Misaligned Plain Bearings—NACA Technical Note 2507: Effect of axial and twisting misalining couples on oil-film pressure distribution of a loaded bearing was determined using 1.62 by 1.62-inch bearings at 5000 rpm shaft speeds with centrally applied loads of 833 psi on projected area plus the misalined loads. Film pressure distribution, in some cases, was found to vary greatly for relatively small load displacements. The 89-page report can be obtained from NACA at the address given above.

NOTEWORTHY

Patents

SCALLOPED CAM SURFACES enable operating pressure to be removed after initially clamping the work in a self-contained collet type work chuck. The cams provide safe, positive means of maintaining the collet grip on the work, and reversely applied pressure is required to release the chuck. Sliding friction between the rotating parts and the operating



yoke is eliminated during rotation, thus reducing the spindle drive load and prolonging the life of the mechanism. Since the customary draw sleeve inside the spindle bore is not required, larger diameter stock can be fed through a given size spindle. Patent 2,-565,330 assigned to Gisholt Machine Co. by Vigo vonKrogh Sundt.

GUIDED BELLOWS SEAL obviates corrosive and leakage conditions at the bonnet of a needle type valve. By enclosing the operative parts in a pressuretight envelope, corrosive fluids may be controlled by the valve without damage to the operating parts, even if they are made of corrodible materials. The valve needle is fastened inside the end cap of the

no housing problem...

Torrington Needle Bearings need no elaborate, expensive housings. A straight bore—machined to proper dimensions—is all that's required. No positioning shoulders... no spacers... no retainers! Simple to install, Needle Bearings are seated by press fit. An arbor press operation does the job easily and quickly.

Reductions in the size—and weight—of housings and related parts are also possible because of the compactness of Needle Bearings. Stronger, more economical designs often result. Want more details? Then drop us a line, today.

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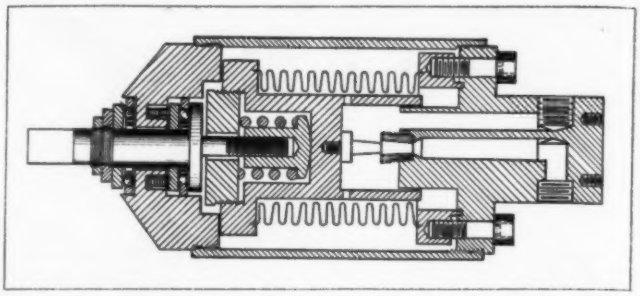
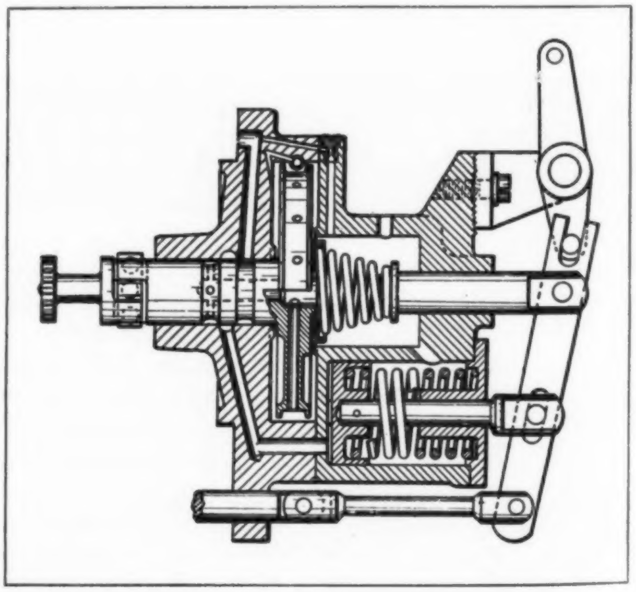
TORRINGTON *NEEDLE* BEARINGS

Needle • Spherical Roller • Tapered Roller • Straight Roller • Ball • Needle Rollers



guided bellows assembly which is telescoped by operation of a screw spindle to regulate the opening at the valve seat. Flexibility of the bellows allows the needle to align itself with the valve seat hole when closing to secure a tight seal. Needle and valve seat are both screw-type replaceable parts. Patent 2,555,996 assigned to United States of America AEC by Rex B. Pontius.

HYDRAULIC JETS are employed to obtain improved speed control in a new engine governor designed to operate from the lubrication system. Driven by the engine camshaft, the jet spinner and control valve are a single unit, the rotation providing high sensitivity in the valve during operation. Hydraulic pressure created by the spinning jets actuates the



control valve regulating the engine oil presure supply to the servomotor piston which adjusts the throttle setting for predetermined engine speed. In the event of engine lubrication system failure, spring reset of the servomotor piston causes the throttle to close, stopping the engine. Patent 2,567,753 assigned to American Bosch Corp., by Heraclio Alfaro.

SEALED ROCKER SHAFT construction to prevent leakage of fluid from within a chamber, where a lever arm must operate through a wall opening, is

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★ *Specified because*

- They are less costly than circular products fabricated by other methods.
- Flash butt-welding produces consistent high-quality welds which are equivalent in strength to the parent metal.
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- Made from rolled or extruded ferrous or non-ferrous material, American welded rings are free from blow holes and harmful inclusions.

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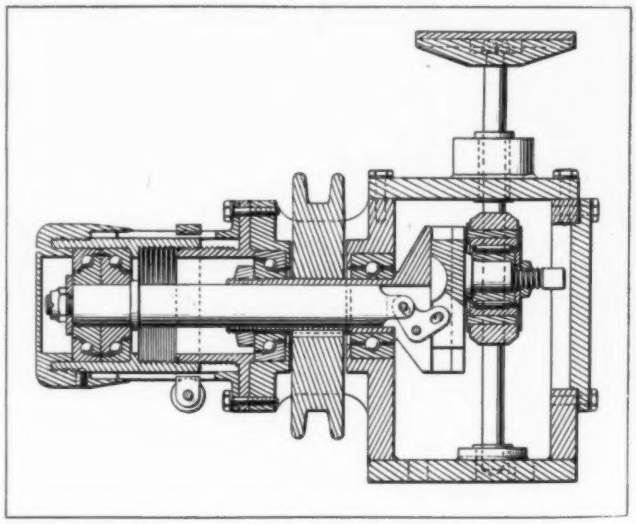
the scope of patent 2,558,103. A split bearing housing is employed to enable assembly of the levers, shaft, bearings, and O-ring seals with the housing cap as a sub-unit. The wall of the reservoir or chamber is machined to receive the cap and lever assembly, and provides the mating half of the bore for the shaft and bearings. The seal rings are located laterally between the cap and chamber openings through which the lever arms extend. This provides an uninterrupted machined bore (except for the cap joint) in which the sealing action is accomplished. The use of multiple seal rings is included in the patent coverage. Inventors Robert E. Ruckstahl and Harold M. Watson have assigned their patent to Westinghouse Electric Corp.

ADJUSTABLE-THROW CRANK enables changing the vibration amplitude of a mechanical vibrator table without stopping the mechanism. The crankpin is adjustable in a radial tee-slot at one end of the crankshaft by means of a screw operated linkage contained inside the shaft. Thrust to position the crank is transmitted to the linkage from a calibrated adjusting knob through a ball bearing assembly. The



crankshaft is belt driven by a separate variable speed unit. Developed for calibrating vibration pickup units used in testing aircraft designs, the invention makes it possible to continuously vary the amplitude as well as the frequency of vibrations during pickup calibration, thus making it possible to more nearly simulate actual dynamic flight-test conditions. Patent 2,569,900 assigned to Glenn L. Martin Co. by R. S. Nevin, Sr. and C. M. Fields.

CLUTCH DRIVE for revolving doors, instead of the conventional stop-and-start electric drive, is covered in patent 2,549,451, assigned to Hudson Bay Co., London, England, by Clement E. Gossling. A motor drive continually rotates a plate, mounted on bearings in the floor, around the center post of the door. Depression of a hand lever on the door causes rollers on the

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No Slipping...High Efficiency...Long Useful Life

● Plant men concerned with preventing production delays and holding machinery operations to top levels have followed the example of leading builders of machinery and equipment. They have seen what the long-life reliability of Diamond Roller Chains, their high efficiency and constant speed ratio, non-slipping operation mean under today's strenuous preparedness production programs.

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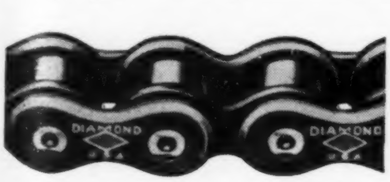
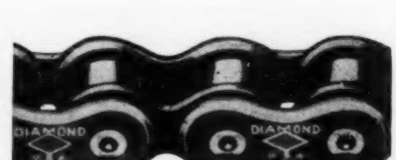
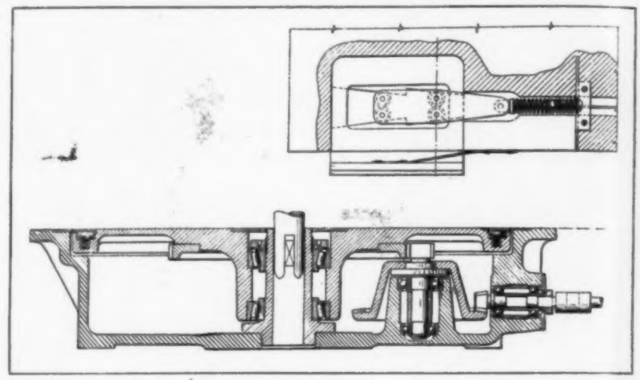


DIAMOND ROLLER CHAINS



clutch control mechanism to move on an inclined track and force the friction surface of the clutch down against the rotating plate in the floor. Release of the



hand lever permits the door to stop. Slope of the roller inclines can be adjusted to permit a small force on the lever to operate the door.

Protective Finishes

(Continued from Page 114)

it lighter" has passed from the air-borne equipment designers to the Quartermaster Corps, ordnance, chemical, military sea transport service and others. The need for a faster moving, more maneuverable military machine has resulted in the specification of aluminum for rocket and shell tanks, rocket motors and other components, portable refrigerators, body armor, shipboard furniture, submarine structures, and a host of other items. That is the reason why the importance of an effective chemical treatment cannot be overemphasized.

Improved Paint Adhesion on Zinc

Where zinc or cadmium coated materiel is painted, zinc-base phosphate coatings are specified. An example of these is Lithoform. Precleaned zinc is acted upon by a brushed, sprayed or immersion bath containing primary zinc phosphate, free phosphoric acid and catalytic agent whereupon there is deposited on the surface a crystalline, zinc-phosphate film highly adsorptive to paint. Paint films ordinarily will not stay on zinc during weathering because of a reaction between them and fatty or other acids in the paint vehicle. The inert phosphate coating prevents this saponification reaction.

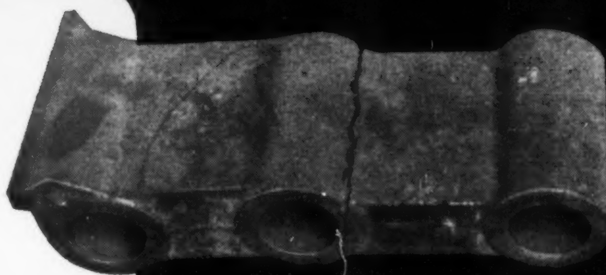
Paint, irrespective of its quality or method of application, is pervious to the atmosphere which contains, among other things, moisture, oxygen and carbon dioxide. These elements filter slowly through the paint, react with the zinc to form oxides and carbonates which in turn produce a chemical soap film between the paint and the metal. This breaks the bond holding the paint, which is then free to peel off under the influence of the normal expansion and contraction of the metal with temperature changes. Sooner or later, depending on the severity of the climate, the exposed protective zinc coating is chemically attacked and destroyed by the elements, leaving the un-

Case History M-109

Here's what we mean by SUPERIOR ENGINEERED FOUNDRY PRODUCTS...

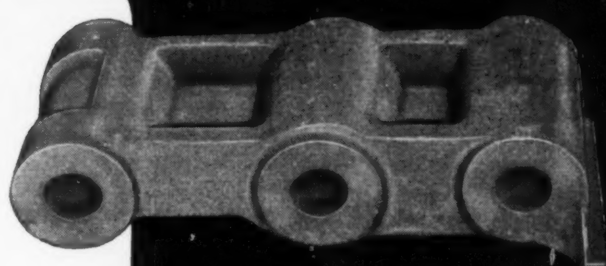
PROBLEM:

1. As originally designed, this cast steel knuckle arm on a die casting machine failed frequently in service, causing costly production delays.
2. Excessive massiveness of part rendered it too inflexible to readily absorb normal operating stresses and strains.



OUR SOLUTION:

A FOUNDRY ENGINEERED DESIGN steel casting in which sections are blended to compensate for high stress areas, thus utilizing metal to obtain structural efficiency. The uniform sections would simplify molding and make it easier to secure a sound casting.



RESULT: 19.3% SAVINGS

1. Elimination of premature service failures and resultant production delays.
2. Massiveness (weight) reduced 28 lbs. while improving reserve strength of part.

TOTAL COST OF PART REDUCED 19.3%

YOU, TOO, CAN GET SAVINGS LIKE THESE! CONSULT OUR PRODUCT DEVELOPMENT SECTION REGARDING YOUR PROBLEM ... WHILE IT'S STILL ON THE DRAWING BOARD

Let our foundry engineers help you conserve critical materials.

ENGINEERED FOUNDRY PRODUCTS

Since 1916

SUPERIOR STEEL AND MALLEABLE CASTINGS CO.

BENTON HARBOR, MICHIGAN, U. S. A.

TO KEEP YOUR CASTINGS COMING ... KEEP YOUR SCRAP GOING TO THE FOUNDRIES

protected iron or steel free to rust.

A dense, stone-like coating of zinc phosphate that is not affected by the atmosphere, that will not chip off, and that forms a firm and lasting bond with paint is produced. As a result, the peeling of all types of paint, lacquer and enamel is practically prevented. Zinc and cadmium surfaces including galvanized iron, galvanneal, cadmium-plated steel, and zinc-base diecastings can be effectively phosphate coated prior to painting.

Processing in a five-stage spray washer for cleaning, rinsing and zinc-phosphate coating is ideally suited to the processing of either hot-dip or electrogalvanized metal rapidly and economically. Therefore, this spray process is the most logical one to use for coating sheet, coiled strip, or duplicate products best processed on a conveyor.

An immersion process used in tanks develops a paint-bonding coating on zinc and cadmium products that have been suitably freed from oil and rinsed. Diecastings, cabinets and miscellaneous products fabricated from galvanized iron are typical of the varied products which are best processed by immersion.

Uses for Coatings

Typical military applications of zinc-phosphate coating chemical include: electrodeposited cadmium plating on various ordnance materiel items; electric power equipment for naval shipboard use; equipment hardware; electronic equipment and component parts of vessels of the United States Navy.

Joint Army-Navy Specification JAN-F-495, entitled *Finishes For Equipment Hardware,* includes the following information.

Type II: Finish shall be enamel conforming to the requirements for Type II of Joint Army-Navy Specification JAN-E-480. Type II finish shall be a protective baked coating suitable for application on zinc or zinc-coated ferrous metals.

PREPARATION of hardware items for finishes: To provide a satisfactory base for the enamel, all hardware, both steel and zinc, shall be given a satisfactory uniform phosphate coating or chemical pretreatment which will further serve as a corrosion inhibitor.

USE: The finishing processes covered by this specification are intended primarily for the treatment of small hardware items such as buckles, clips, snaps, etc., manufactured for attachment to equipment. Unless otherwise provided for by specifications, contracts or orders, the processes are not intended for hardware for buildings, boats, automotive equipment, mechanical equipment for buildings, fire apparatus, or portable or prefabricated buildings, nor for equipment which is to be furnished as a unit.

Type II finish is intended for use where protection from corrosion is of major importance. This finish should always be used on steel equipment hardware that is a part of web or canvas equipment.

Finishes covered by this specification should not be used on surfaces which will be in frequent or constant contact with food products, except where such use is sanctioned by competent medical authorities as approved by the agency or bureau concerned.

Part 2 of this series will discuss rustproofing, and protecting friction surfaces with phosphate coatings.

REPORT ON

Materials

Latest forecasts on the materials situation indicate that there will be little relaxation of stringent materials controls, at least in metals, before mid-1952 or 1953. Assuming that the mobilization schedule continues as at present, supplies of steel and aluminum may ease by this summer, but most critical shortages will continue until at least 1953.

The worst metals shortages will be in copper, which may continue to be in tight supply indefinitely. The Defense Production Administration has reported that copper supplies are "almost dangerously short"—and very little relief is in sight.

New production facilities now being built will probably ease the aluminum situation after midyear. By the end of this year the nation's primary aluminum capacity will be approximately 60 per cent greater than the 1950 level (about 40-45 per cent more than last year's rate), according to the Aluminum Association. Reactivation of World War II plants now shut down, and additional plant capacity now contracted for but not yet built, will boost the 1953 output above this figure.

New steel mills, scheduled to start rolling in late 1952, will increase the supply of steel plates. Structural shapes will continue to be scarce until at least 1953. Steel sheets, most used in metalworking, should ease after this July.

Boron Steels

Use of boron-treated low-alloy and carbon steels, recently publicized as substitutes for higher-alloy steels, has increased appreciably, according to the NPA. Output in November was forecast at 37,000 tons, compared with 35,000 tons in October.

Low-nickel chromium-molybdenum steel has been one of the most popular of the new steels. About 7500 tons of this boron-treated steel are being used per month for carburized gears.

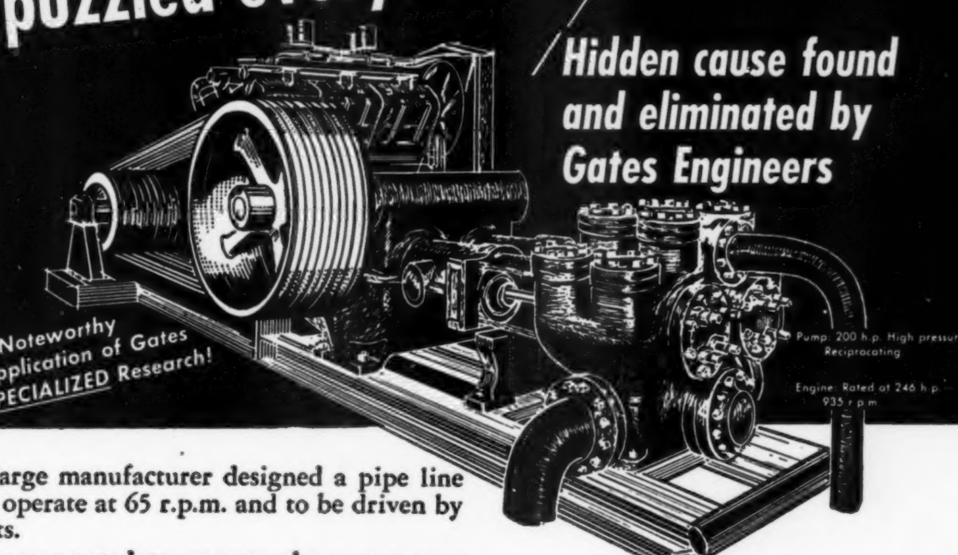
Plywood Substitutes for Metals

Possible uses of plywood as a substitute for scarce metal were discussed recently at a conference between the Department of Defense and industry spokesmen. As an example, the new Unicel plywood freight car, now undergoing performance tests, was cited.

Principal barrier to the proper use

Severe vibration on pump drive puzzled everyone. . .

Hidden cause found and eliminated by Gates Engineers



A Noteworthy Application of Gates SPECIALIZED Research!

A large manufacturer designed a pipe line pump to operate at 65 r.p.m. and to be driven by 12 V-Belts.

A user wanted to operate the pump at a speed of 90 r.p.m. and realized that additional horsepower would be required. Upon checking the V-Belt drive, it was found that the additional belts required would increase the face width of the sheave and therefore the moment arm of the bearing load. Any increase in bearing load combined with the increased speed was distinctly undesirable and therefore the user sought some means of reducing the overhung weight and width of the sheave.

Learning that a *special* V-Belt had a sufficiently higher horsepower rating to gain the greater pump speed with *only 9 belts,* he had the drive re-designed to use these belts.

When installed on the job, the belts vibrated severely. At a speed somewhat below the desired range, the belts would fly completely off the sheaves. Operation was so rough that it was almost impossible to run the pump at all.

Gates Engineers were called in for consultation. Their analysis indicated that the particular belts used formed a *too-compliant* link between the pump and the engine. This allowed tortional vibration to build up to excess. Therefore, belts of *less compliance* would have to be used.

Accordingly, Gates Engineers wrote specifications for belts having the necessary h.p. rating but with the exact *degree of compliance* that would avert the vibration. These special V-Belts were built by Gates—installed—and the drive operates smoothly under all loads and at all speeds within the range.

The excessive vibration is gone. The clutch operates smoothly at any speed. And 13 additional pumps have since been equipped with identical belts—all operating most satisfactorily.

Gates ability to analyze and correct obscure drive difficulties is no accident—for Gates operates the largest V-Belt Testing Laboratories in the world. And Laboratory findings are carefully checked by tests made under actual field conditions. Finally, the *results* of these exhaustive tests are immediately reduced to *usable data* for the design of V-Belt drives to *perform whatever task may be required.*

Phone for a Gates Field Engineer

Only under exceptional circumstances, of course, will you ever need belts of special construction. But, more often, some drive in your plant may not be operating quite as it should. Or you may have a particularly difficult drive to design. Again, you may want to be sure what size and construction of V-Belts will give the most efficient and the lowest cost service on a certain drive. In any case, you have only to phone a Gates Field Engineer, *always near you* in all industrial centers.

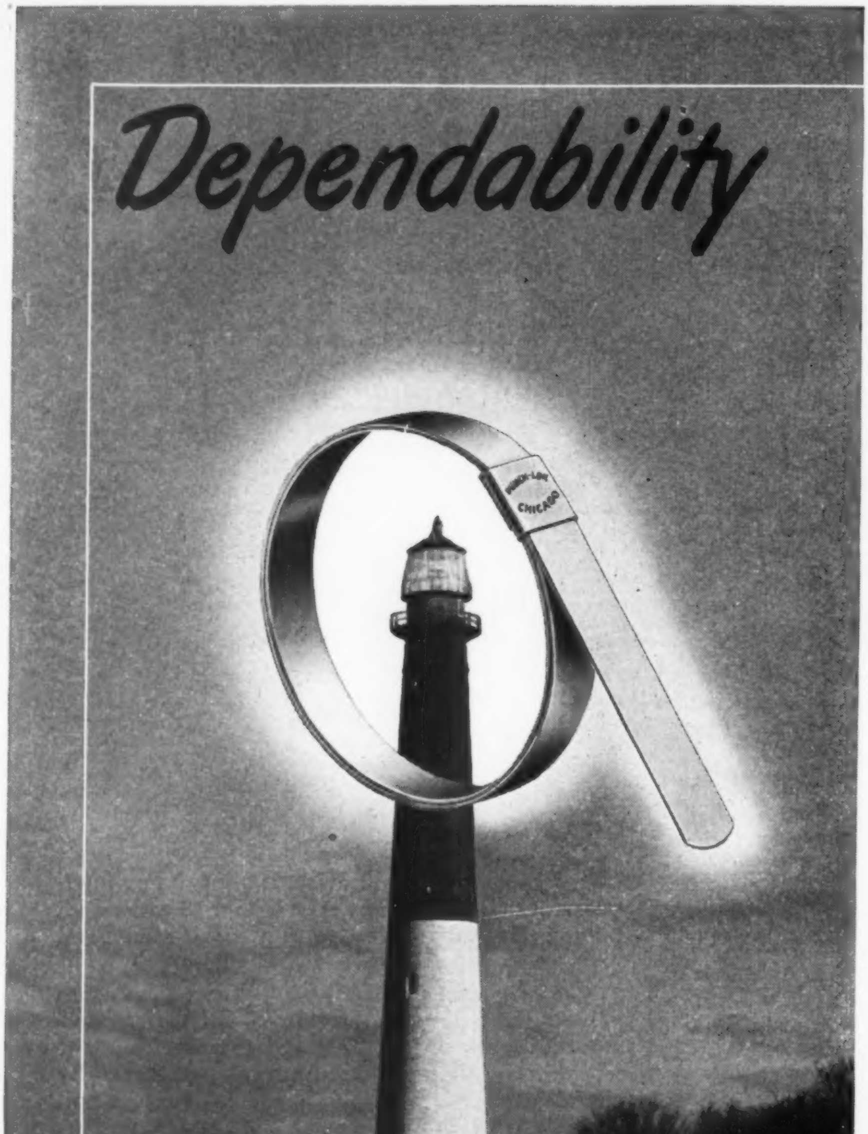
Just look in your phone book under "Gates Rubber." A Gates Field Engineer will come right to your plant and put at your service the full benefits of Gates V-Belt knowledge and experience *without the slightest obligation!*

ENG-518

Gates
Hose V-Belts
Molded Rubber Goods
for Industry

VULCO ROPE DRIVES
ENGINEERING OFFICES AND JOBBER STOCKS
IN ALL INDUSTRIAL CENTERS

THE GATES RUBBER COMPANY
The World's Largest Makers of V-Belts
DENVER, U. S. A.



of plywood is lack of understanding of the various grades of plywood available. The better grades of plywood are now in restricted supply, but abundant amounts of the lower grades, well suited to many applications, are available.

Tungsten and Molybdenum Allocations

The United States' share of the free world's tungsten and molybdenum supply during the third quarter of 1951 was almost half of all tungsten and three-quarters of all molybdenum produced. Despite a rising production during the third quarter, total free-world demand for tungsten and molybdenum (excluding U. S. stockpile requirements) exceeded supply by more than 80 and 70 per cent respectively.

Substitute Finishes

Most electroplated finishes for civilian use have gone, or are scheduled to go, out of existence. Nickel, tin and cadmium plates are no longer available. Chrome plate needs an intermediate nickel plate, and is of dubious value without it. Possible substitutes are:

1. Black oxide coatings on aluminum, formed by anodizing the work in sulfuric acid baths and then dyeing the oxide coating
2. Amorphous aluminum - chromium phosphate coatings (giving 500 hours of salt spray resistance with specified paint systems)
3. A single-dip chromate conversion coating, developed for wrought and cast aluminum.

Also suggested (by the DPA) as substitutes for chrome and nickel plating are bright-zinc plating, extra-fine aluminum lining pigment, synthetic enamel resembling chrome, and aluminum-base baking enamel.

Phosphate coatings, high in corrosion resistance and providing a good bond for organic finishes, have been specified by defense agencies for many military applications. Descriptions of these coatings are given on Page 108.

List of Administrators

Questions regarding the application of provisions of NPA M-Orders, Regulations and CMP Regulations should be referred to the proper administrator for the particular order or regulation. Although Department of Commerce district and regional offices can answer many questions regarding materials controls, very often detailed interpretations are needed—and can be supplied only by the appropriate administrator listed in the following.

NPA REGULATIONS: The basic rules of the NPA priority system are given by five regulations. Materials

for production purposes are covered in three of these regulations. Administered by the Policy Co-ordination Bureau of the National Production Authority, these regulations, with names and addresses of the administrators*, are:

1. Inventory Control
   Nathan B. Salant, Production Evaluation Div., 2U8 NG*
2. Basic Rules of the Priorities System
   W. C. Groce, Priorities & Directives Div., 235 OG
5. Appeals....T. C. Mumford Boyd, Appeals Board, 5800 Comm.

CMP REGULATIONS: The Controlled Materials Plan has certain basic rules of its own, also administered by the Policy Co-ordination Bureau of the NPA. Of the seven basic regulations, four are of interest to production-materials users. These are:

1. Basic Rules of the Controlled Materials Plan
   John F. Skillman, Technical Coordination Div., 3K2 NG
3. Same—Preference Status of Delivery Orders
   John F. Skillman, Technical Coordination Div., 3K2 NG
4. Deliveries of Controlled Materials by Distributors
   John F. Skillman, Technical Coordination Div., 3K2 NG
2. Inventories of Controlled Materials
   Nathan B. Salant, Production Evaluation Div., 2U8 NG

M-ORDERS: The following NPA M-orders are directly concerned with materials. The following list is partial, but represents the M-orders covering materials most used by designers. Division and Bureau addresses are given.

Metals and Minerals Bureau

IRON AND STEEL DIV.

M-1: Iron and Steel........K. H. Hunter, 2E7 NG; Herbert Johnson, 2G7 NG

M-24: Tin Plate and Terneplate .... F. A. McLelland, 206 NG

M-80: Sched. A: Nickel-Bearing Stainless Steel, High Nickel Alloy and Nickel Silver..W. J. McCune, 2A6 NG

Sched. B: Tool Steel and High Speed Steels ....Felix Kremp, 2A6 NG

M-80: Sched. 1: Nickel .....Harold Larson, 2K9 NG

Sched. 2: Cobalt ....F. F. Franklin, 2L8 NG

Sched. 3: Tungsten ..F. F. Franklin, 2L8 NG

Sched. 4: Molybdenum .... F. F. Franklin, 2L8 NG

Sched. 5: Columbium and Tantalum F. F. Franklin, 2L8 NG

M-81: Pure Tungsten and Pure Molybdenum H. M. Lusk, 2L9 NG

ALUMINUM AND MAGNESIUM DIV.

M-5 Rated Orders for Aluminum .... T. A. Lynch, 2N1 NG

M-84: Aluminum for Destructive Uses John West,, 2J2 NG

TIN, LEAD AND ZINC DIV.

M-8: Tin ....W. L. Raup, 2F1 NG

M-9: Distribution of Zinc ..John Sellon. 2G1 NG

M-15: Use of Zinc .... Mrs. Margaret B. Murphy, 2G2 NG

M-19: Cadmium .... Mrs. Margaret B. Murphy, 2G2 NG

M-38: Lead ....F. J. Owen, 2H1 NG

M-39: Antimony .....F. J. Owen, 2H1 NG

M-48: Bismuth ......F. J. Owen, 2H1 NG

M-76: Distribution of Lead ...F. J. Owen, 2H1 NG

* Full addresses, according to key given, are shown at end of section.

COPPER DIV.

M-11: Copper and Copper-Base Alloys Frank H. Hayes, 2H15 NG

M-86: Copper Wire Mill Distribution.. Paul Andrews, 2H13 NG; W. E. Bradford, 2H15, NG

MISCELLANEOUS METALS AND MINERALS DIV.

M-54: Platinum ...H. B. Sharpe, 2A2 NG

M-66: Artificial Graphite and Carbon Electrodes ....Atwood B. Oatman, 2B3 NG

Chemical, Rubber and Forest Products Bureau

RUBBER DIV.

M-2: Rubber ....E. D. Kelly, 2H15 NG

CHEMICAL DIV.

M-32: Chemicals (Limitation for DO Rated Orders ....John Gosnell, 2Y14 NG

Textile, Leather and Specialty Equipment Bureau

LEATHER AND LEATHER PRODUCTS DIV.

M-28: Leather ..Charles B. Floyd, 2154 TT

LUMBER AND LUMBER PRODUCTS DIV.

M-63: Softwood, Plywood .... Richard D. Behm, 2N7 NG

Keyed addresses given in the above listing are as follows:

NG—New GAO Bldg., Washington, 25, D.C.
OG—Old GAO Bldg., Washington, 25, D.C.
TT—Tempo T Bldg., Washington 25, D.C.
Comm—Dept. of Commerce Bldg., Washington 25, D. C.

A complete listing of administrators, or further information on regulations and M-orders, can be obtained from district and regional offices of the Dept. of Commerce.

Design Abstracts

(*Continued from Page* 162)

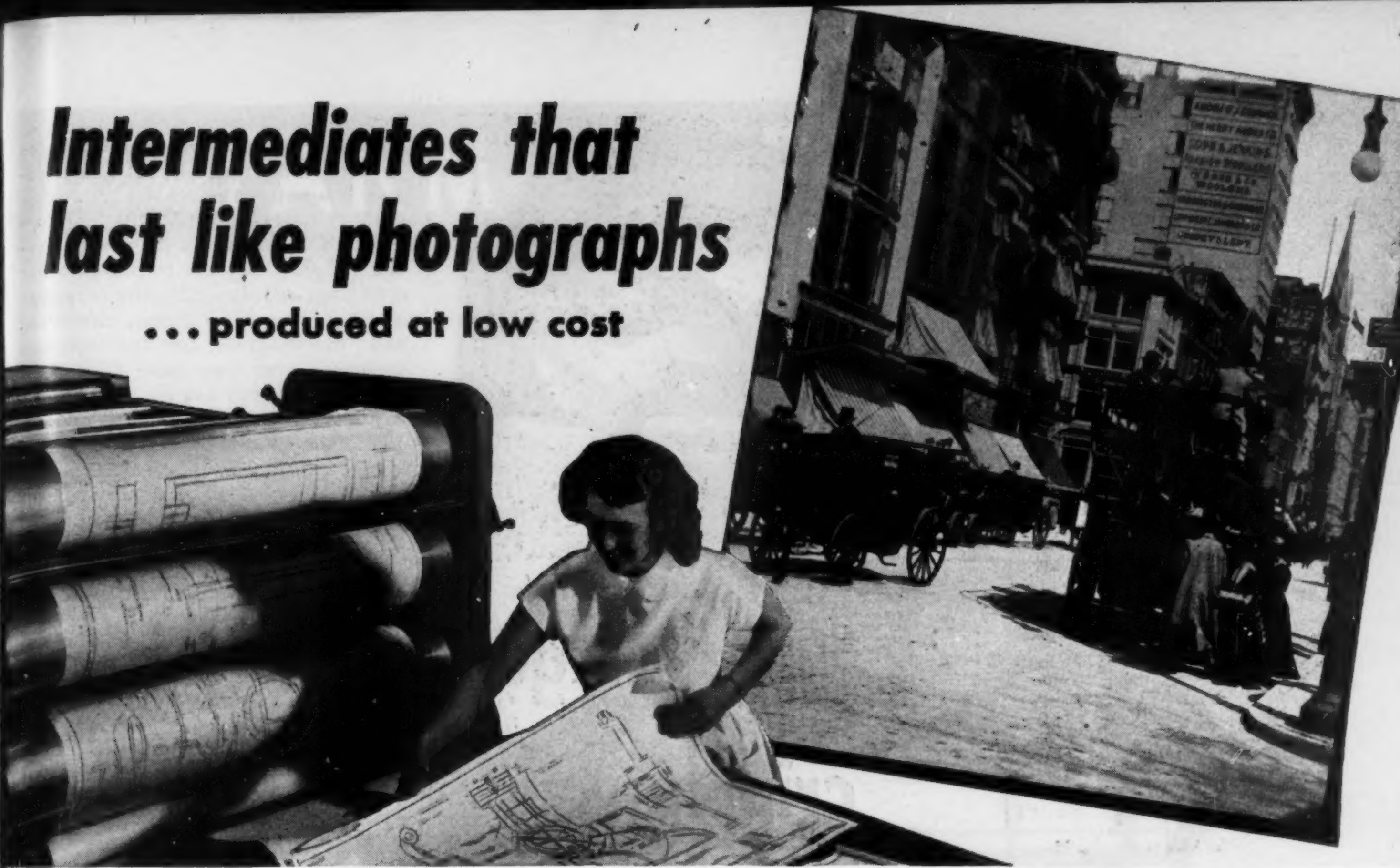
mills it has been found that boron steels can be rolled and forged in the same manner as steels of like composition without boron provided the boron content is below the level at which hot shortness occurs. It has been found that the type of scale which forms on boron steels is rather light and flaky, is easily removed and has little tendency to adhere to the steel and be driven into the surface in forging or other hot forming operations. Because of their lower total alloy content boron steels are thought to be less sensitive to shatter cracks than the more highly alloyed steel. This fact simplifies heating and controlled cooling practices. The people that have used these steels have found that their machinability is comparable to the base analysis without boron.

The formability characteristics of boron steels in both hot and cold working have been found to be similar to the characteristics exhibited by the base analysis without boron. Generally speaking, this has made possible improved fabricating practices and higher production in terms of units per hour.

When heat treated to the same hardness values and the same condi-

Intermediates that last like photographs

...produced at low cost



A case history based on the experience of the Hyster Company, Portland, Oregon

By reproducing its engineering drawings on Kodagraph Autopositive Paper, the Hyster Company gets intermediates which have dense black photographic lines on a translucent, highly durable paper base. *Intermediates* which will remain intact in the files year after year . . . and produce sharp, legible blueprints and direct-process prints whenever needed.

And the cost is surprisingly low because Kodagraph Autopositive Paper boasts unique photographic properties which cut production costs *substantially*. It can be handled in ordinary room light . . . and it produces positive copies *directly*—without a negative step.

Furthermore, existing equipment can be utilized—Autopositive can be exposed in any blueprint or direct-process machine ...and processed in standard photographic solutions. Hyster uses a blueprint machine which has also been designed to produce Autopositive intermediates in a continuous flow.

How The Hyster Company Uses Autopositive

For All Print Production. Autopositive intermediates are used instead of valuable and perishable original drawings. Producing sharp, legible blueprints and direct-process prints time after time, they greatly reduce the possibility of reading errors in the shop.

For "Permanent" Files in its home office . . . and branches in the United States and abroad. "Photolasting" Autopositive intermediates eliminate the costly delays experienced in the past when intermediates faded, turned yellow, or became brittle.

For Drafting Room Short Cuts. Autopositive Paper is used to reclaim old and weak-lined drawings . . . and to duplicate standard designs which will be incorporated in new drawings. *All of which saves hours of creative drafting time.*

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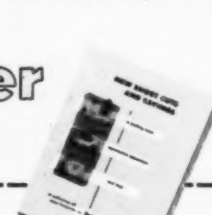

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Gentlemen: Please send me a copy of "New Short Cuts and Savings"... describing the many savings Kodagraph Autopositive Paper is bringing to industry.

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tion of finish as conventional alloy steels boron steels have exhibited like or better endurance limits and abrasion characteristics.

It is necessary that the development of boron steels be pushed forward rapidly in order to conserve our dwindling supply of the more versatile alloys such as nickel, chromium and molybdenum for the complex technical jobs they have to do and which boron will not do. If our "alloy civilization" is to develop and expand, the limited known reserves of the conventional alloying elements must be saved for more important uses than conferring hardenability on engineering steels.

From a paper presented at AISI regional meetings in Philadelphia and Cleveland, 1951.

Why Supply System Diagrams?

By J. P. Bellafaire
Chevrolet Detroit Gear & Axle Div.
General Motors Corp.
Detroit, Mich.

A DESIGNER and fabricator of a machine, having lived with the machine since its inception, sometimes loses sight of the fact that what information he takes for granted and does not record, is of vital necessity to the user of the machine. This information permits a user to rapidly familiarize himself with the machine, especially with the part usually considered the most mysterious—the electrical control system.

The problems facing production and maintenance personnel are unchanged whether they exist in the manufacturing or processing of automobiles, button holes, clothing, shoes, milk, butter, or candy. Manufacturing processes require highly co-ordinated and economically loaded machines to meet the scheduled production. These machines have become complex and much of the production skill, heretofore under the control of the operator, has been transferred to electrical, hydraulic, pneumatic, and mechanical controls. Instead of the operator furnishing the skill and the mental processing required in the production of a part, these functions are built into the machine through the use of motor starters, contactors, relays, solenoids controlling both pneumatic or hydraulic valves, push-buttons, limit-switches, timers, counters and the other control accessories. As more skills and processes are transferred into the machine from the operator, the more highly complex become the machine controls and with the modern trend the operator merely supplies

parts into the machine for processing.

Down time of a machine is costly. Money invested in a nonproductive machine or paid to an idle operator must be eventually absorbed, in a higher unit cost of production. Therefore, returning the machine and the operator to production in the minimum of time is of vital importance.

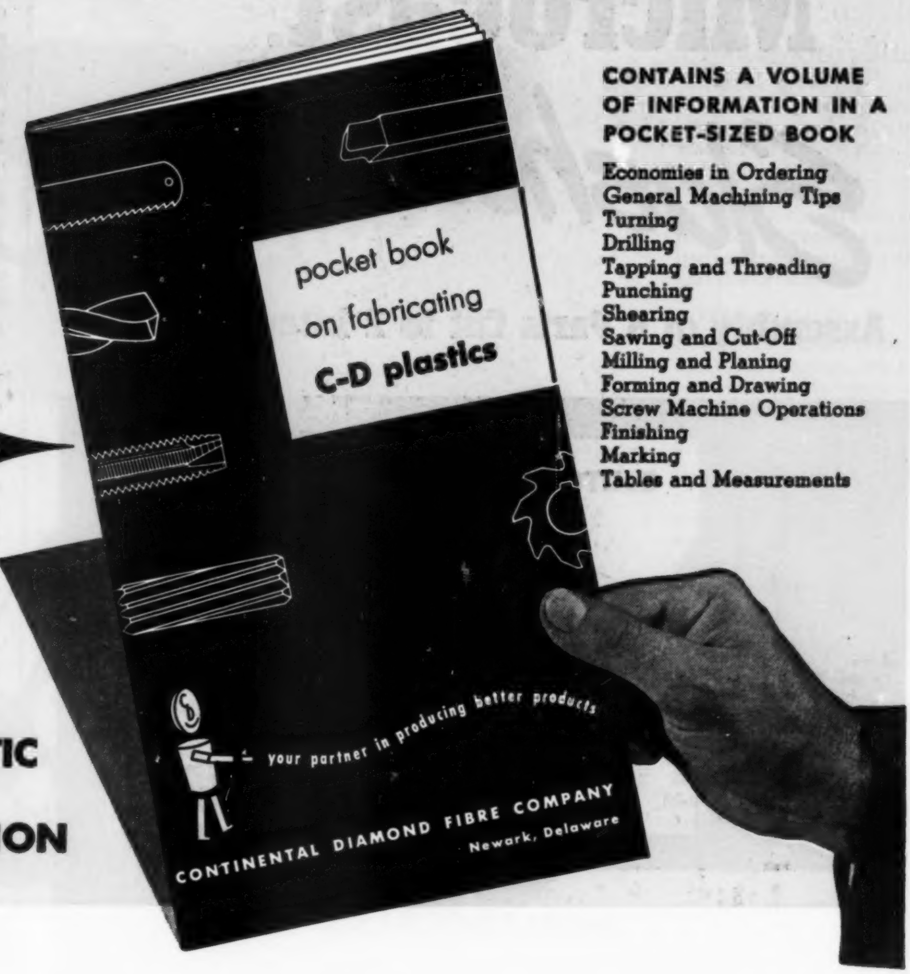
The maintenance man is faced with the problem of quickly finding any trouble. The description of the erratic condition given to him by the machine operator, job setter, or production foreman must be related to a controlled function of the machine and to the factors affecting that function. A limit switch may have failed to operate or a solenoid may have stuck open or burned out. The problem is in locating the particular limit switch or solenoid.

The complete wiring diagram, available at the machine, has proven to be most effective in minimizing down time. This complete wiring diagram is in a form that contains the electrical schematic diagram, sequence of operation, schematic layout of the machine components such as air and hydraulic equipment associated with the control of the machine, and a bill of material—all the necesary information on one sheet. Thus, instead of paging over prints for the complete wiring diagram, the maintenance man can rapidly acquaint himself with the machine as he glances from the schematic diagram to the sequence of operation and the schematic machine layout.

Use Standard Symbols

The schematic, elementary or line diagram, showing in simple form how the machine is wired and what the power and control circuits consist of, should use the standard graphical symbols for electrical devices. Brief explanations are marked beside the device symbol to indicate the condition of a control device as related to machine functions. A limit switch contact symbol with a brief explanation indicating its condition when actuated by, or as related to, a machine component has been found to give a fuller understanding of its function in the control circuit. For example, beside a limit switch contact would be the brief description: *LS 7—NO—Closed end of spindle forward stroke.* The range of pressure settings controlling a pressure switch indicated on the diagram eliminates questions concerning its characteristic or function. Words describing what solenoids control in the machine cycle when energized lessen the mystery that sometimes surrounds them. A solenoid symbol with the note, *Sol A—Brings*

NEW



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clamps forward, firmly implants the idea of its functions. Also inrush and sealed values of current for solenoid operation help in determining erratic conditions of the device. A contactor or relay coil symbol may have a short description of the cycle function that it directly initiates, such as, *Returns back-up cylinder*. Of great importance on the schematic line diagram are the terminal point markings or numbers, since they enable the maintenance man to make a continuity check or trace the connections of the devices and locate the source of trouble on the machine.

The sequence of operations should be brief and simple, yet adequate enough to convey the cycle progression. Lengthy and involved sentences of description have a tendency to confuse; however, limit-switch actuations, timer functions and push-button operations should be adequately explained. If pressure switches are required in the circuit, the settings at which they function and how pressure variations are introduced should be indicated. When selector or drum switches are incorporated, a brief description of how their various settings affect the machine cycle eliminates guess work of functions affected.

Completeness Important

A limit switch and solenoid schedule placed at the bottom of the sequence of operation aids the maintenance man in identifying their function in a few minutes. It helps him to readily localize the trouble and provides a key for locating each limit switch on the machine. In hydraulically or pneumatically operated machines a solenoid schedule serves as a key to the control of the movements of the machine.

The complete wiring diagram includes a schematic layout of the machine components showing the physical relation of the moving machine parts to the control or limit switches which they act upon during the machine cycle. It presents a picture which allows quick visualization of the control scheme of the machine while the sequence of operation is reviewed and the line diagram analyzed.

To complete the wiring diagram a bill of material is included on the same sheet. Every electrical device used in the machine control should be listed in enough detail to permit its replacement if found defective. The manufacturer's name, type or model, and catalog number are important. For example, contactors or relays should be listed according to ratings and the number of normally open and normally closed contacts. Control devices should be iden-

EVERY **SEALMASTER** BEARING

Pre-Lubricated

FOR A LONG STRETCH OF SERVICE

1. PERMANENTLY SEALED
Felt-lined steel flinger, rotating in labyrinth, excludes dirt and retains proper amount of lubricant.

2. SELF-ALIGNING
Bearing unit, with seals independent of housing, can align itself in any direction without seal distortion.

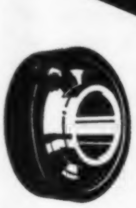
3. PRE-LUBRICATED
Before shipment from the SealMaster factory, the bearing chamber is filled with proper amount of lubricant.

4. NO HOUSING WEAR
Patented locking pin and dimple prevent rotation of outer race in housing. This eliminates housing wear, permits shaft alignment and positions unit for relubrication.

5. FLOATING RETAINER
Ball retainer is designed to float on ground inner surface of outer race. Traps lubricant, prevents churning.

6. HOUSING DESIGN
SealMaster's cast housings combine rigid, one-piece construction with smooth streamlined appearance.

Pillow block



Cartridge unit

Flange unit

Flange-cartridge unit

Take-up unit

Bearing maintenance at low cost is a feature worth designing into your machines. In SEALMASTERS the lubricant is sealed-in at the factory. Thus, SEALMASTER Ball Bearing Units assure smooth operation, and long life—with reduced power consumption.

Ask for Catalog 845—for complete data on SEALMASTER *Bearings.*

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A DIVISION OF STEPHENS-ADAMSON MFG. CO.

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tified in the bill of material with the same numbers or letters that are used in both the line diagram and sequence of operation. Often, when this is omitted, the maintenance man is forced to consult the diagram, when replacing the device, in order to determine its electrical characteristics.

From a paper entitled "Information Required for Trouble Shooting on Machine Tools" presented at the AIEE Fourth Annual Machine Tool Conference in Rockford, Ill., November 14-16, 1951.

High-Speed Roller Bearings

By E. F. Macks, Z. N. Nemeth and W. J. Anderson

Lewis Flight Propulsion Laboratory
Cleveland, Ohio

THE rolling contact bearing is used in present-day turbojet engines because it has certain advantages over the hydrodynamic bearing for this application. These advantages are, primarily, its lower starting torque and its comparative insensitivity to oil interruptions.

With respect to bearing surface speed for both the rolling contact and hydrodynamic bearing types, the present maximum turbojet service is about 220 feet per second. The maximum service under consideration by designers is about 500 fps. The region of present knowledge for the rolling contact bearing reaches about 120 fps and for the hydrodynamic bearing it extends to about 210 fps.

The turbine wheel of a turbojet engine is generally supported on a roller bearing to allow for thermal expansion of the shaft in the axial direction. For any bearing to function, it must operate at an equilibrium temperature below the tempering temperature of the bearing steel and below the breakdown temperature of the lubricant. Therefore, a safe balance must be maintained between the heat input and the heat removed. The operating temperature of the bearing has therefore been chosen as the overall criterion of bearing operation.

Heat is generated within the bearing by friction at the bearing surfaces which are in sliding and rolling contact as well as by churning of the oil by the moving parts. Heat also enters the bearing by conduction through the shaft, by conduction through the housing, and by radiation and convection from the surrounding environment.

To maintain a safe equilibrium bearing operating temperature this


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"bearing heat" must be removed. For high speed bearings the lubricant removes a major part of this heat. The function of the lubricant has thus been extended to include cooling, as well as lubricating. In the research to be discussed the study has been limited to the lubrication and cooling of conventional types of roller bearings in which the heating of the bearing occurs only by the friction and churning within the bearing.

Bearing speed, bearing load, and cage design have been investigated since these variables affect the heat generation within the bearing. Oil flow and oil inlet temperature, velocity, viscosity, and distribution have also been investigated because these variables affect the heat removal from the bearing.

Test Procedure

The test bearing was mounted on one end of a test shaft which was supported in cantilever fashion so that component parts of the bearing and the lubricant flow could be observed during the operation. Radial load was applied to the test bearing by means of a lever and dead-weight system in such a manner that the outer race of the test bearing was essentially unaffected by small shaft deflections or by small shaft and load-arm misalignments.

The test bearings were 75-mm bore, 130-mm outside diameter cylindrical roller bearings (size 215) of the type currently used as the turbine roller bearing of turbojet engines. The bearings were equipped with one-piece inner race riding bronze cages. Five oils of widely different viscosities were used in the investigation.

Effect of Variables

The following discussion is an analysis of the different variables in terms of their effects on bearing operating temperature.

Speed: The bearing temperature increases approximately linearly with an increase in speed over the operable range of bearing surface speeds (about 80-350 fps). If the bearing surface speed is increased beyond a critical value of about 350 fps the bearing will not reach an equilibrium operating temperature. This indicates an incipient bearing failure. These high speed failures usually occur as surface damage at the cage locating surface in presently designed high-speed roller bearings.

Load: The effect of load on bearing operating temperature for both an inner-race riding and a roller-

riding cage-type bearing was determined at a surface speed of 290 fps. There is little change in bearing temperature with a decrease in load down to about 300 lb. Also, there is little difference in the operating temperature of each bearing type in this load range. Below 300-lb load, however, the temperature of the roller-riding cage-type bearing decreases rapidly as contrasted to the temperature decrease of the inner-race-riding cage-type bearing. The difference in the operating characteristics between the cage-type bearings investigated can be explained on the basis of the different amounts of heat generation at the cage contacting surfaces as influenced by the roller slip between the rollers and the inner race.

The low load region is of interest since the shaft stiffening bearings of certain turbojet engines operate at very light loads and high speeds. Considerable service trouble at this bearing location has occurred due to raceway scuffing. While this trouble has been encountered with both bearing types, the roller-riding cage-type bearing has inherent advantages for this particular operating condition due to its lower operating temperature. This type bearing therefore better lends itself to development for the solution of this particular bearing problem. It is of interest to note that the outer-race-riding cage-type bearing will also show a reduction of heat generated at the cage locating surface as slip increases.

Curves were plotted for roller slip as influenced by speed, load, and cage type. The general trend of these curves indicates an increase in slip with increase in speed, and a decrease in slip with increase in load.

Sources of Friction

In a bearing having a roller-riding cage only one source of cage friction exists, and that is between the cage pockets and rollers. In a bearing having an inner-race-riding cage, however, sliding friction occurs at the surfaces between the cage and the inner race, as well as in the roller pockets. However, these two bearing types have about the same operating temperatures under certain equivalent operating conditions over about 300-lb load. This indicates that the heat generated in the cage pockets of the roller-riding cage-type bearing is about equal to the sum of the heat generated in the cage pockets and at the cage locating surfaces of the inner-race-riding cage-type bearing.

This apparent paradox is due to the fact that the loads in the cage pockets of the roller-riding cage-type

bearing are greater than are the loads in the cage pockets of the inner-race-riding cage-type bearing, since the cage of the roller-riding cage-type bearing not only spaces the rollers but also supports itself at these surfaces. However, in the case of the inner-race-riding cage-type bearing the cage only spaces the rollers in the roller pockets, while supporting itself on the inner race flanges.

For each bearing type at light loads under about 300 lb the same amount of roller slip occurs between the rollers and the inner race. Also, for each bearing type the roller speeds about their respective axes will be the same. For each bearing, then, the cage supporting load has decreased about the same amount and the relative speed in the cage pockets has also decreased by the same amount. In the inner-race-riding cage-type bearing, however, there is the additional heat generated at the cage locating surface—for with roller slip the relative speed increases at this surface since the shaft speed remains constant while the cage speed decreases. It is believed that this source of cage frictional heat accounts for the major part of the difference in the operating temperature of these two bearing types in the low-load high-speed region of operation.

Temperature Control

Oil Flow: The bearing operating temperatures were found to be minimum when the single oil jet used for these tests was directed perpendicular to the bearing face, that is, not at an angle into or with the direction of shaft rotation. The bearing temperatures were also found to be minimum when the oil was directed at the cage locating surface rather than at the cage or at the space between the cage and the outer race. There is an appreciable decrease in bearing temperature with an increase in oil flow. This decrease in bearing temperature results so long as complete oil scavenging is maintained—the oil must not be allowed to collect around the bearing and churn into a froth. Over the flow range where churning is allowed to occur without immediate removal of the churned oil there may be an increase in bearing temperature with an increase in flow. This is a concept not entirely appreciated by many users of high-speed bearings.

Oil Velocity: For a constant oil flow of 4 lb per minute there is a decrease in bearing inner-race temperature with a decrease in oil jet diameter. This effect is due to the

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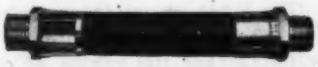

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increase in oil inlet velocity associated with the decrease in oil jet diameter at a given oil flow. This increased oil stream velocity results in an increase in the film coefficient of heat transfer due both to the velocity effect and to the fact that the higher velocity stream is better able to penetrate the windage barrier set up by the high-speed bearing components.

Oil Inlet Temperature: Bearing operating temperature decreases approximately linearly with a decrease in oil inlet temperature. This reduction in bearing operating temperature is appreciable. For example, for each 1 F decrease in oil inlet temperature the test bearing temperature decreased 0.5 F to 1 F depending upon the operating condition.

Oil Inlet Viscosity: For the oils used in the investigation, at inlet temperatures of 100 F and 250 F the bearing operating temperature is higher for a higher viscosity oil at constant flow; therefore, a higher flow of a higher viscosity oil is required to maintain a constant bearing temperature. To maintain a constant bearing temperature the oil flow is approximately a linear function of the oil inlet viscosity at the inlet temperature over the range of viscosities investigated when the data is plotted on semilog paper.

Oil Inlet Distribution: The number of the oil jets, as well as their location, influence the distribution of oil to the bearing and therefore serve also to control the bearing operating temperature.

Load Position Influential

It has been found that no appreciable difference in maximum bearing operating temperature occurs when a single jet is indexed circumferentially with respect to a unidirectional load acting on the bearing. The loads investigated were 368 and 1120 lb. Furthermore, it has been found that the region of outer-race-maximum temperature always occurs 60 deg to 120 deg before the oil-jet location in the direction of shaft rotation and the region of minimum outer-race-maximum temperature always occurs 60 to 120 deg after the oil-jet location in the direction of shaft rotation regardless of the relative location of the oil jet with respect to the load vector.

An evaluation was made of the cooling efficiencies of various oil introduction techniques from comparison between a single oil jet, a radial hole in the outer race, and 12 jets. The 12 jets were spaced circumferentially about one side of the bearing. For a given oil flow the 12 jets

system resulted in appreciably lower bearing inner-race temperatures than did the other systems. Also, as one might expect, the temperature variation about the outer race is much less for the 12-jet system. This circumferential variation was found to be but 5 F for the 12 jet system, compared to 93 F for the outer race hole and 35 F for the single jet. An outer race circumferential temperature variation may cause an out-of-round of the outer race which might adversely influence the bearing performance and its reliability.

Engine Tests Confirm Data

A series of tests were completed with the rear turbine bearing of a turbojet engine instrumented to determine if the foregoing correlation also applied to engine bearing data. Inner race temperatures as well as outer race temperatures were obtained. The engine correlation is fair.

The cooling curve for the engine data lies slightly above the cooling curve for the test rig data and has a lesser slope. The differences in level and slope of the outer-race cooling curves for the engine and test rig data are due to the different heat flow paths to and from the engine and test rig bearings, and particularly to the external heat associated with engine operation.

Conclusions: The test rig results as well as the turbojet engine data have been generalized by means of dimensional analysis (shown in original paper) to allow the designer to predict the bearing temperature rise from a single curve regardless of whether bearing speed, bearing load, oil flow, oil inlet temperature, oil inlet velocity, or oil inlet viscosity vary over wide ranges.

From paper No. 51-*A*-66, *"Operating Characteristics of Cylindrical Roller Bearings at High Speeds" presented at the ASME Annual Meeting in Atlantic City, N. J., November* 25-30, 1951. *Complete copies may be obtained from ASME,* 29 *W.* 39*th St., New York* 18; $0.25 *each to members,* $0.50 *to nonmembers.*

Annual Index

The 1951 Annual Index to the editorial contents of MACHINE DESIGN, which was published in the December issue, Page 284, has been reprinted. If you need an extra copy you may obtain one free of charge by writing to the editorial department.

News

OF MANUFACTURERS

ENTERING the aviation industry through a newly organized California subsidiary, **Crane Co.** of Chicago has acquired the assets formerly owned by Hydro-Aire, Burbank, Calif. The latter company produces aircraft valves, filters, actuators and other high precision hydraulic, pneumatic and electric accessories.

•

Caterpillar Tractor Co., Peoria, Ill., has announced an expansion program that will include a new factory in York, Pa., extensive additions to the Joliet, Ill. plant, and some improvements in the Peoria plant. The plant being built at York will cover approximately 360,000 sq ft and will be devoted to the production of track and track-roller parts.

•

The stock holders of Lithgow Corp., applicators of Lithcote protective baked phenolic coatings and linings, recently voted to change the name of the company to **Lithcote Corp.**

•

A new plant designed for the manufacture of Selectron, a polyester resin plastic, is under construction at Springdale, Pa., according to an announcement from the paint division of **Pittsburgh Plate Glass Co.**, Pittsburgh. The new building will adjoin the company's paint-producing plant and is expected to be completed this spring.

•

Spencer Rubber Products Co., Manchester, Conn., manufacturer of phenolic rubber sponge, has entered the industrial molded rubber goods field. The addition of modern toolmaking and molding equipment will enable the company to produce precision molded rubber products in almost any size, shape or form.

•

Completion of a new 56,000 sq ft manufacturing building to be devoted exclusively to small, miscellaneous transmission parts has been announced by the **Fuller Manufacturing Co.**, Kalamazoo, Mich. The new building will house machine lines for such parts as bearing covers, gearshift and hand-brake levers, clutch throw-out yokes, bearing carriers and

shift bar housings for heavy-duty transmissions, etc. The addition will permit the company to devote facilities of the present main plant entirely to the production of gears, shafts, cases and clutch housings, and to the final assembly of transmissions.

•

Auburn Button Works Inc., Auburn, N. Y., custom molder of plastics, has completed a new experimental laboratory for its recently expanded extrusion department. The new laboratory will handle quality control of raw materials and finished products, the development of new vinyl compounds, and research for better production methods.

•

A building program undertaken by the **Shallcross Manufacturing Co.,** Collingdale, Pa., is now nearly completed and includes a new wing on the main factory building for general production facilities, expansion of the firm's instrument laboratory, and the establishment in separate quarters of a large component development laboratory devoted to precision resistors and various electronic specialties.

•

Boston Woven Hose and Rubber Co., Cambridge, Mass., has broken ground on a new $1,000,000 rubber reclamation plant. Using an air separation process developed by company engineers, the new factory will help to conserve rubber supplies by extracting all usable material from scrap automobile tires and other waste rubber. The entire output will go into the company's own manufacturing operations, including government orders. It is expected that the plant will be completed sometime in the second quarter of 1952.

•

A new pump plant is to be constructed in Kansas City, Mo., for **Fairbanks, Morse & Co.,** Chicago. Construction is scheduled to begin in the spring and will provide a machine shop containing 190,000 sq ft of floor space, a foundry with 150,000 sq ft and an office building with 20,000 sq ft.

•

The fifth annual competition of the **Lincoln Arc Welding Foundation's** Engineering Undergraduate Award and Scholarship Program has been announced, and a booklet of rules and conditions is now available from The James F. Lincoln Arc Welding Foundation, Cleveland 17, O. This annual program is a competition for undergraduate engineers to encourage them in developing an engineering project in their own field. All registered un-


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dergraduate engineers are eligible to compete. Awards totaling $6750 are made for the best papers on design of machines or structures, or separate components of machines or structures, in which arc welding is the method of fabrication.

•

Cadillac Plastic Co., has moved into new and larger quarters at 15111 Second Ave., Highland Park, Mich. The new 15,000 sq ft, one-floor building more than doubles the company's former facilities and will house all local activities of the company, including executive, sales and general offices, warehousing and supply, and manufacturing and fabricating.

•

A large addition to the manufacturing facilities of the **Macklin Co.,** Jackson, Mich., manufacturer of grinding wheels, is nearing completion. It will increase production capacity of the present plant by more than 35 per cent. Increased floor space will provide additional facilities to the finishing, inspection, Resinoid-oven and vitrified-kiln departments. A new tunnel kiln designed to accommodate 48-in. diameter wheels is being erected.

•

A new 26,000-sq ft factory expansion was recently completed at Cheboygan, Mich., by **Detroit Tap and Tool Co.,** Detroit.

•

Four employees of **Allegheny Ludlum Steel Corp.,** Pittsburgh, recently were presented with the Allegheny Ludlum Award in recognition of their outstanding achievements on behalf of the company. The awards include a citation from the board of directors, "The President's Medal," and $1000. Schuyler A. Herres won his award for development work on titanium and titanium base alloys and their production; Douglas S. Gormly, for his work in the development of a process for hot topping steel ingots. A joint award to Messrs. I. G. Shoff and John Eagleson was made for their work in processes for pickling high alloy and silicon strip.

•

Elastic Stop Nut Corp. of America, Union, N. J., has announced the signing of an agreement of merger with the American Gas Accumulator Co., Elizabeth, N. J. The latter company's business will be carried on as a division of Elastic. AGA operates plants at Elizabeth, Chicago, and Toronto, Canada. Its principal products include aviation ground-lighting equipment, aids to marine navigation, Agastat time-delay relays used in electrically controlled circuits.

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A new, 28 page catalog has just been released by the Ortman-Miller Machine Co. which gives complete engineering specifications, data on O-M's special internal locking system and special full listing of O-M parts. Prepared for designers and users of cylinders for any application, it covers standard, oversize and 2-1 piston rods, giving full information on all sizes from 1½" to 8" bores.

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Society

ACTIVITIES

The **American Society for Testing Materials** has announced the establishment of an annual H. W. Gillett Memorial Lecture, the first to be given at the society's 50th anniversary meeting in New York City during the week of June 23, 1952. The purpose of this lecture, which is being sponsored in co-operation with Battelle Memorial Institute, is to commemorate Horace W. Gillett, who was one of America's leading engineers and metallurgists, the first Director of Battelle in Columbus, O., and an active worker for many years in ASTM.

Readers will recall Dr. Gillett's interesting and thought-provoking articles in MACHINE DESIGN.

♦

Wellwood E. Beall, Boeing Airplane Co., has been elected president of the **Institute of the Aeronautical Sciences** for 1952. Newly elected vice presidents are: Preston R. Bassett, Sperry Gyroscope Co.; Smith J. De France, NACA-Ames Aeronautical Laboratory; Edward H. Heinemann, Douglas Aircraft Co. Inc.; and John C. Leslie, Pan American World Airways System. Charles H. Colvin, G. M. Giannini & Co. Inc., was elected treasurer. Re-elected were: S. Paul Johnston as Director of the Institute, Robert R. Dexter as Secretary, and Joseph J. Maitan as Controller.

♦

Announcement has been made by the **Resistance Welder Manufacturers Association** of its annual prize paper contest which offers a total of $2,250 in prizes for outstanding papers dealing with resistance welding subjects. The awards will be made at the 1952 fall meeting of the society. Complete details regarding the subject matter of the papers and other contest rules may be obtained from the Resistance Welder Manufacturers Association, 1900 Arch St., Philadelphia 3, Pa.

♦

Elected at the **Gray Iron Founders' Society's** annual meeting in Chicago were the following officers and directors: President, E. L. Roth, Motor Castings Co.; vice president, R. G. Schaefer, Schaefer-Goodnow Foundries Inc.; secretary, H. P. Good, Textile Machine Works; and treasurer, H. J. Trenkamp, The Ohio Foundry Co. New members elected to the board of directors were: T. I. Curtin Jr.,

Waltham Foundry Co.; W. A. Morley, Link-Belt Co.; Max Kuniansky, Lynchburg Foundry Co.; C. H. Ker, Dalton Foundries Inc.; J. E. Quest, J. F. Quest Foundry Co.; and E. P. Trout, Lufkin Foundry & Machine Co. D. H. Workman was reappointed executive vice president, and C. O. Burgess was appointed technical director for the ensuing year.

♦

The **American Society of Mechanical Engineers** presented the following awards at its 1951 Annual Meeting, Chalfonte-Haddon Hall Hotel, Atlantic City, N. J.

Hoover Medal to William L. Batt, minister in charge of the Economic Co-operation Administration mission to the United Kingdom, in recognition of his "leadership in engineering, management, and public responsibility, and his many distinguished services to his community and the nation."

John Fritz Medal to Ervin G. Bailey, Babcock and Wilcox Co., for "outstanding engineering achievements in the field of combustion and distinguished service to his fellows in advancing the engineering profession."

The *Hoover Medal* and *John Fritz Medal* are joint awards of the four "founder societies," ASME, AIMME, ASCE, and AIEE.

Henry Laurence Gantt Medal to Thomas Roy Jones, Daystrom Inc., for "distinguished achievement in industrial management as a service to the community." This medal is an annual award of ASME and AMA.

ASME Medal, highest award of the society, to Glenn B. Warren, General Electric Co., for "leadership in the science and art of turbine design."

Holley Medal to George Rupert Fink, National Steel Corp., for "unique acts of courageous engineering vision."

Worcester Reed Warner Medal to Professor Jacob P. Den Hartog, Massachusetts Institute of Technology, for "outstanding contributions to engineering literature."

Richards Memorial Award to J. Kenneth Salisbury, General Electric Co., for "outstanding achievement in mechanical engineering within 20 to 25 years after graduation."

Melville Prize Medal to Clayton H. Barnard, Bailey Meter Co., for the "best original paper or thesis on any mechanical engineering subject presented before the ASME the previous year."

Gold Medal of Pi Tau Sigma, honorary engineering fraternity, to Assistant Professor Warren M. Rohsenow, Massachusetts Institute of Technology, for "outstanding achieve-

ment in mechanical engineering within ten years after graduation."

Junior Award of the society to John D. Stanitz, NACA, for his paper "Analysis of the Exhaust Process in Four-Stroke Reciprocating Engines."

Honorary memberships conferred on: Robert M. Gates, Air Preheater Corp.; Professor Lewis F. Moody, Princeton University; Latham E. Osborne, Westinghouse Electric Corp.; and Harry A. Winne, General Electric Co.

Elected to serve with Reginald J. S. Pigott, the new **ASME** president, were the following regional vice presidents: Willis F. Thompson, Westcott & Mapes Inc.; Ernest H. Hanhart, consulting mechanical engineer; Ernest S. Theiss, Davey Compressor Co.; and Samuel H. Graf, Oregon State College Engineering Experimental Station. New directors-at-large are: Albert C. Pasini, The Detroit Edison Co.; and Paul B. Eaton, Lafayette College mechanical engineering department.

♦

At its Nineteenth Annual Meeting, **The Packaging Machinery Manufacturers Institute** elected the following officers for the coming year: President, G. Radcliffe Stevens, Elgin Manufacturing Co.; first vice president, Palmer J. Lathrop, Cameron Machine Co.; second vice president, Edwin H. Schmitz, Standard-Knapp Div., Emhart Manufacturing Co. Three new directors elected were: Mrs. Helen Horix Fairbanks, Horix Manufacturing Co.; Edwin E. Messmer, Amsco Packaging Machinery Inc.; and Herbert H. Weber, H. G. Weber & Co. Inc.

♦

Charles S. Craigmile, Belden Mfg. Co., was elected president of **National Metal Trades Association** at the 52nd Annual Convention of the association held in the Blackstone Hotel, Chicago. Others elected to office were: Earle S. Day, Collyer Insulated Wire Co., first vice president; and Norman L. Rowe, Ideal Roller and Mfg. Co., second vice president and treasurer.

♦

The Engineering Manpower Commission of **Engineers Joint Council** announced that over 400 local representatives have been designated in all 48 states, as well as Canada and the District of Columbia, for the purpose of "beating the drum" for fullest industrial and military utilization of engineers, increased enrollment of students in engineering colleges, and fullest draft board consideration for this category of needed manpower.

SALES AND SERVICE

Personnel

RECENTLY appointed assistant sales manager of the Arrowhead Rubber Co., Downey, Calif., **Arthur W. McGuire** will be concerned with sales problems in connection with the company's line of products, including O-rings, silicone rubber items and Airtron fiberglass ducting being produced at the new Long Beach plant. Prior to his new appointment Mr. McGuire was sales manager of the aircraft division of Chicago Metal Hose Corp. and was associated with the industrial sales division of the Crane Co., Oakland, Calif., for several years.

♦

Three appointments in the sales organization of the General Electric Co. specialty transformer and ballast department, Fort Wayne, Ind., were announced recently. **H. K. Pritchard** has been named manager of general purpose transformer sales; **J. P. Coughlin**, manager of aircraft and electronic transformer sales; and **A. E. Rowe**, manager of lighting component sales.

♦

James S. Anderson, who has been on leave from The Babcock & Wilcox Tube Co., Beaver Falls, Pa., has returned to his duties as assistant general sales manager. He has been in Washington serving as chief of the Tubing Section of the Iron and Steel Division of the Industries Operations Bureau of the National Production Authority since April 1951.

♦

Kennametal Inc., Latrobe, Pa., recently announced the following appointments: **Kenneth Trombley** has been assigned to the new Tennessee and Alabama sales district, with headquarters at 18 Clearview Ave., Chattanooga, Tenn.; and **Frank Price** has been appointed engineer and representative in the middle Atlantic district.

♦

W. G. Turner, formerly manager of the southeastern region, with headquarters at Atlanta, has been transferred to Cleveland as manager of the Great Lakes region for Cummins Engine Co. Inc., Columbus, Ind. New manager of the southeastern region, **R. P. Parshall**, was manager of the Milwaukee branch of Cummins Diesel Sales Corp. The company also recently promoted **R. F. Davis**, former assistant manager of the central re-

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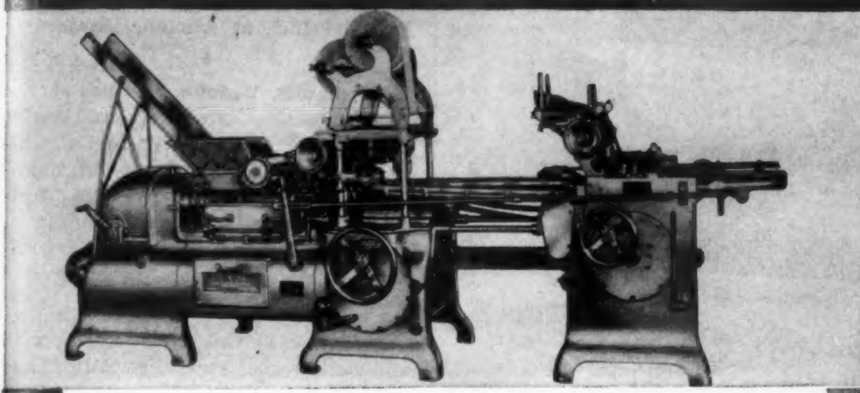

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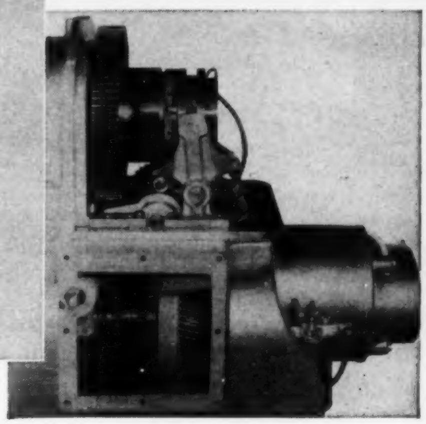
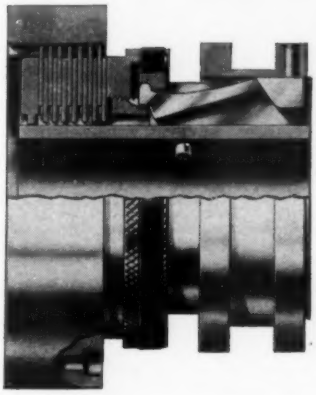


THE INSTALLATION



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For six years the Lynch Corporation, Packaging Machine Division, Toledo, Ohio, has used two Maxitorq floating disc Clutches in this Morpac butter print forming, wrapping and cartoning machine.

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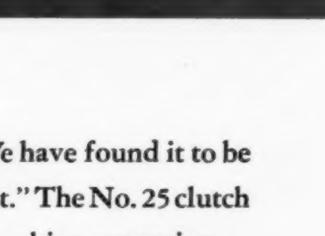

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gion, headquartered at Chicago, to manager of the eastern region, with offices in the Chrysler Bldg., New York City. **Walter N. Westland,** former manager of the eastern region, now heads Cummins Diesel of New England Inc., at Allston, Mass.

♦

Norman B. Weinke, who has served as production manager and sales manager of Cullman Wheel Co., Chicago sprocket and roller chain manufacturer, has been elected vice president. In his new capacity he is in charge of an expanded sales organization, as well as manufacturing.

♦

A. Milne & Co., New York, solid and hollow tool steel distributor, has appointed **F. J. Grant,** formerly Boston branch manager, to the position of New England sales manager. His headquarters are at 631 East First St., Boston 27, Mass. The company has also appointed **F. A. Bade** as representative in the state of Kentucky, with headquarters at 51 Hill Rd., Louisville, Ky.

♦

Edward K. Vaughan has been transferred from the Chicago staff of The New Jersey Zinc Sales Co. to the headquarters office in New York City, and **Robert L. Campbell,** who has been associated with the Chicago office in sales work with the pigment division, has been transferred to the metal division to replace Mr. Vaughan.

♦

Quaker Rubber Corp., division of H. K. Porter Co. Inc., Philadelphia, has appointed **J. J. Merkel** and **E. E. Klemm** as branch managers of the Detroit and Cleveland districts, respectively. Both men have been associated with the company in sales capacities for several years.

♦

Homer L. Lacock, 907 South Ave., Rochester 20, N. Y, has been appointed district field engineer in charge of the northern New York territory, representing the Dayton Rogers Mfg. Co., Minneapolis, on its complete line of activities.

♦

The appointment of **A. J. McAllister** as president and general manager of the Detroit Gear division of Borg-Warner Corp. has been announced. Mr. McAllister succeeds **Howard E. Blood,** who has been president of the division since 1923, before it was merged with Borg-Warner. Mr. Blood will continue as a vice president and director of the parent corporation and will also assume charge of the new Products Develop-

ment Laboratory which is being established in Detroit for the purpose of developing new and improved types of automatic transmissions and other products.

♦

Reeves Pulley Co., Columbus, Ind., manufacturer of variable-speed control equipment, has appointed **James A. Miller** as manager of its newly opened sales and engineering office in Los Angeles. He will be assisted by **Harlan M. Gillis**, from the engineering department of the Reeves home office. Mr. Gillis served for seven years in the New England office.

♦

Alfred Lippman Jr. has been named general manager of the Commonwealth Engineering Company of Ohio, Dayton, O.

♦

The appointment of **L. J. Smith** to the newly created position of eastern regional sales manager has been announced by the Chiksan Co., Brea, Calif. His headquarters will be at the company's Newark, N. J., office, where he has served as a field engineer for the past four years.

♦

E. E. Alexander was recently placed in charge of the Thomson Electric Welder Co. office in Cleveland. He has long been associated with the company's home office staff in Lynn, Mass.

♦

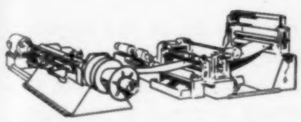
A new assignment of responsibility has been made in the operating management of the Eddystone plant of Baldwin-Lima-Hamilton Corp., Philadelphia. Vice presidents **John S. Newton, Raymond B. Crean** and **James R. Weaver** will have complete responsibily for the locomotive division, the Southwark division and the Special Products division, respectively.

♦

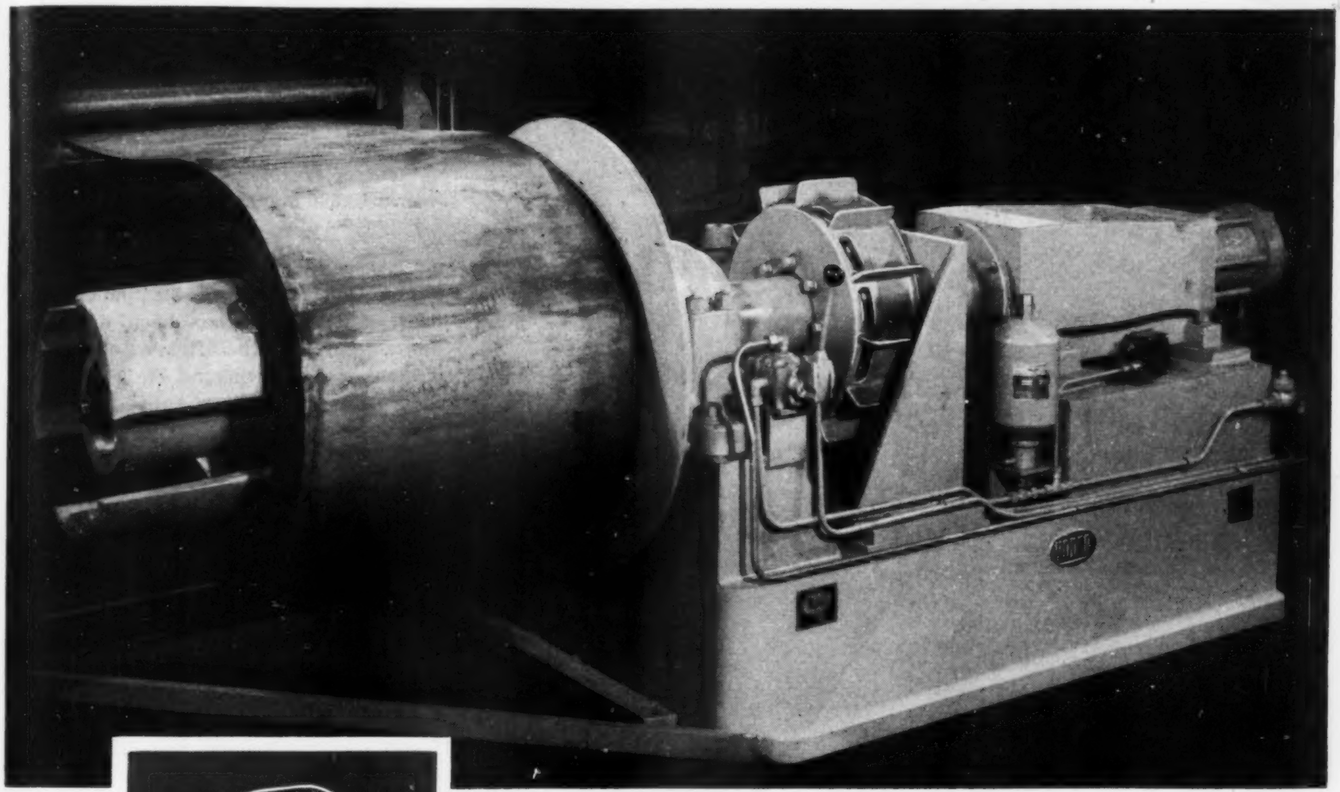
Appointment of **H. Barden Allison** as district sales manager of the Philadelphia branch, mechanical goods division, United States Rubber Co., has been announced. He succeeds **A. B. Means**, who will continue in the capacity of sales advisor.

♦

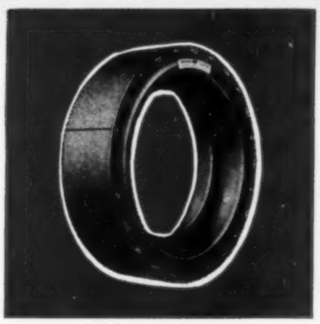
With headquarters in the Maritime Bldg., 911 Western Ave., Seattle, Wash., **Harry W. Gordon** has been appointed northwest district manager for The American Pulley Co., Philadelphia. His territory includes the states of Washington, Oregon, Montana, Idaho and Wyoming and the provinces of British Columbia and Alberta, Canada. At the same time



TYPICAL EXAMPLE OF FAWICK BRAKE APPLICATIONS IN MODERN PROCESSING MACHINES



Uncoiler of Yoder Slitting Line with Fawick "E" Brake on mandrel shaft. The Line slits 36" mild steel in gages up to .014" with as many as 15 cuts, at speeds up to 330 FPM. Maximum back tension on line is 638 pounds, total pull.



FAWICK "E" Brake with sensitive, positive control of internal-expanding brake action and 360° full-width friction surface.

FAWICK Ventilated Brake Drum with internal and external ventilating fins for maximum dissipation of operating heat under heavy-duty use.

The FAWICK "E" VENTILATED BRAKE used on the Yoder Slitting Line contributes several important operating advantages to this high-efficiency equipment.

The Fawick Brake is incorporated in the uncoiler unit to maintain back tension on the strip being recoiled. This is accomplished by a continuous "drag" action of the Brake with proper tension maintained by increasing or decreasing air pressure in the brake actuating tube.

The brake design provides remote control from the master control panel at the slitter section where the operator can instantly regulate tension to work requirements.

Operating heat caused by continuous slip action of the brake is minimized by the internal and external ventilating fins of the Fawick "E" Brake Drum.

Fawick design and operating advantages that contribute to high operating efficiency and long, trouble-free life on thousands of "performance-proved" installations can be profitably used to solve your Clutch and Brake problems.

FAWICK AIRFLEX CO., INC.
9919 CLINTON ROAD • CLEVELAND 11, OHIO

FAWICK Airflex
INDUSTRIAL CLUTCHES AND BRAKES

For further information on Fawick Industrial Clutch and Brake Units, write to the Main Office, Cleveland Ohio, for Bulletin ML-22.

the company announced the appointment of **Sidney H. Hewett** as district manager in Detroit, to cover the territory including sections of Michigan, Ohio, Indiana and Kentucky. His headquarters are at 1357 Sheridan Ave., Plymouth, Mich.

Whitney Chain Co., Hartford, Conn., has announced the appointment of **Harold W. Beder Jr.** as general sales manager.

Associated with the Stackpole Carbon Co., St. Marys, Pa., since 1929, **H. A. Williams** has been appointed manager of the firm's electronics components division. He has served as sales manager of this division for a number of years.

G. H. Baumann was recently appointed sales manager for Feedrail Corp., New York.

The Timken Roller Bearing Co. has announced the appointment of **Sherman R. Lyle** of its Cleveland office to the position of district manager of the steel and tube division, northern Pennsylvania and New York State district.

Having served as a project engineer since June 1951, **William L. Parlon** has been named general manager of Elbeeco Inc., wholly owned subsidiary of Aeroquip Corp., Jackson, Mich.

Benjamin Sampson, sales manager, has been elected vice president of the K. H. Huppert Co., Chicago, manufacturer of electric furnaces and ovens. He will continue to supervise the sales of the company as well as co-ordinate sales, engineering and manufacturing.

Charles H. Besly & Co., Beloit, Wis., has named **Roger A. Kenman** as sales manager of the machine tool division.

J. T. Bell has been promoted to Detroit district manager for the Mid-West Abrasive Co., Owosso, Mich. Concurrently, **James J. Corcoran** was placed in charge of the customer service division, and **D. F. McDonald** and **G. Reagh Atkinson** were appointed as service engineers.

New York Belting and Packing Co., Passaic, N. J., has appointed **Donald C. Howard** as factory representative in California, Nevada and southern Oregon.

No Cost Barriers Now

to PRECISION QUALITY

Fine-Pitch Gears to tolerances in the "low tenths" are produced economically on Fellows equipment. Production records of the Fellows 3-Inch Fine-Pitch Gear Shaper and its companion No. 4 Fine-Pitch Gear Shaving Machine furnish the proof. These machines, together with the precision accuracy of Fellows cutters and shaving tools, make possible the mass production of small fine-pitch gears to very close limits. The 48 pitch gear shown above was held to 0.0005" after cutting and 0.0002" after shaving. If you have a fine-pitch gear production problem, we can help you. Call or write the nearest Fellows Office.

Fellows No. 4 Fine-Pitch Gear Shaving Machine

Fellows 3-Inch Fine-Pitch Gear Shaper

Fellows

THE FELLOWS GEAR SHAPER COMPANY

Head Office and Export Dept. Springfield, Vermont, Branch Offices: 323 Fisher Bldg., Detroit 2; 5835 West North Avenue, Chicago 39; 2206 Empire State Bldg., New York 1.

DESIGNERS

Everyone concerned with the design and application of involute gears will want a copy of the 1950 edition of "The Involute Curve and Involute Gearing". Write the nearest office on your business letterhead.

GEAR SHAPERS · SHAVING MACHINES · CUTTERS AND SHAVING TOOLS · GEAR INSPECTION INSTRUMENTS · THREAD GENERATORS · PLASTICS MOLDING MACHINES

S A L E S

Notes

FORMATION and organization of Acme Steel Products Division of **Acme Steel Co., Chicago,** has been announced. The new division will operate and function as an independent sales and distributing company for steel strapping, tools and accessories, stitching wire and equipment, as well as other related industrial strip steel products. Sales of strip steel and special products will remain with the parent company, as will all manufacturing and production operations. President of the new division is John C. Bucuss, former general manager of the strapping division and a veteran of 33 years of service with Acme.

♦

The Oliver H. Van Horn Co. Inc., 1742 St. Charles Ave., New Orleans 1, La., has been appointed representative to handle distribution and sales engineering in Louisiana and Mississippi on broaching machines, broaches, broach sharpening equipment, etc., produced by Colonial Broach Co., Detroit. The Van Horn Co. has branches in Baton Rouge and Shreveport, La., and Houston, Tex.

♦

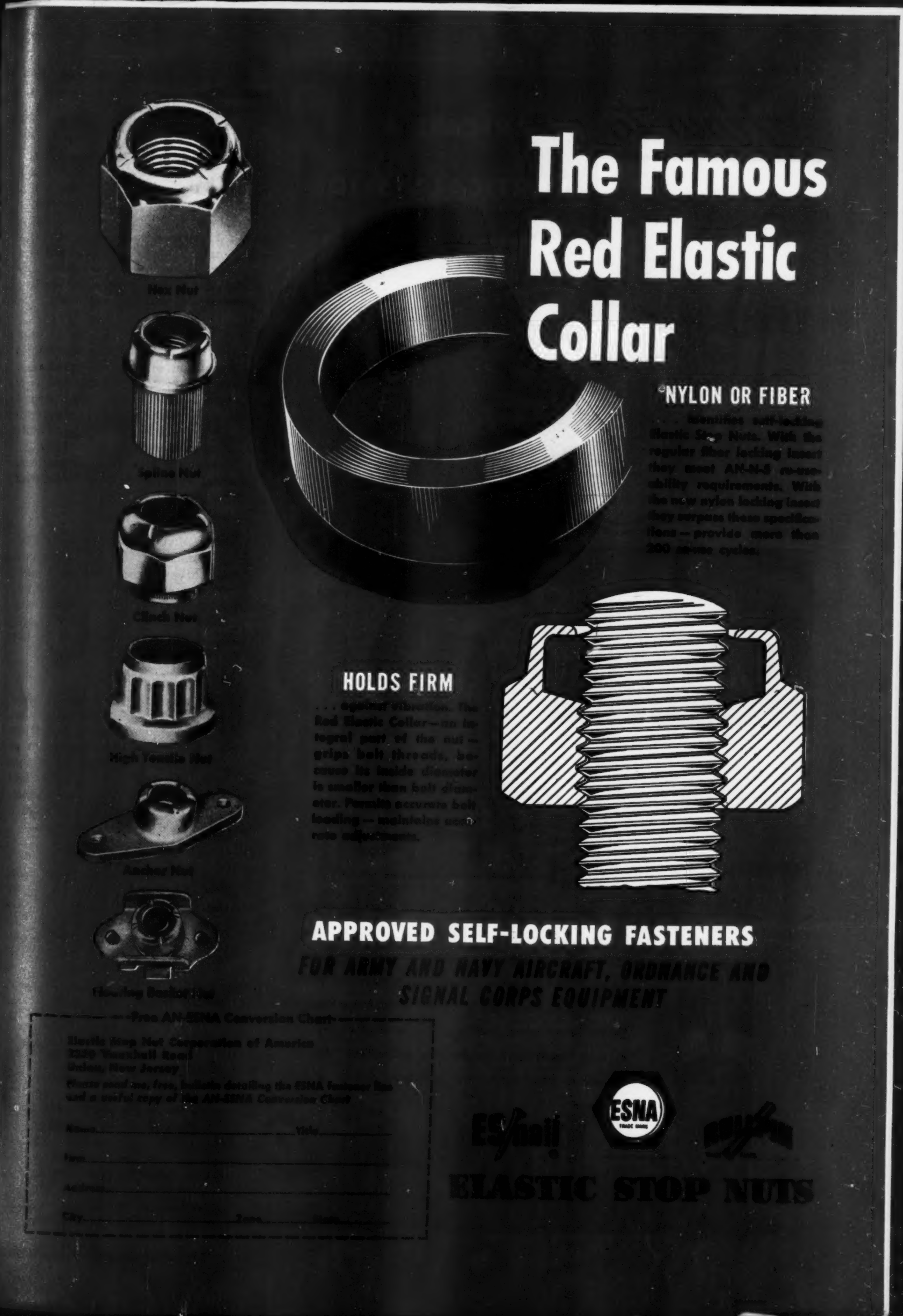
A new district office, located in the Kline Village development, was opened recently at **Harrisburg, Pa.,** by **Minneapolis-Honeywell Regulator Co.**

♦

In order to provide specialized service in the sale and development of felt and fibrous products, **Armstrong Cork Co., Lancaster, Pa.,** has established a separate felt and fibrous products department of the Industrial Division. D. P. Paiste, formerly manager of the Industrial Division district office at Boston, has been named manager of the new department.

♦

A full-length, full-color industrial motion picture covering virtually all phases of modern steel foundry operations and techniques for the production of carbon steel, special alloy and stainless steel castings, has been completed by **Lebanon Steel Foundry,** Lebanon, Pa. Entitled "Steel with a Thousand Qualities," the film is being made available free of charge for showing by and to industrial and manufacturing groups, technical, engi-



The Famous Red Elastic Collar
NYLON OR FIBER
. . . identifies self-locking Elastic Stop Nuts. With the regular fiber locking insert they meet AN-N-5 re-usability requirements. With the new nylon locking insert they surpass these specifications — provide more than 200 re-use cycles.
HOLDS FIRM
. . . against vibration. The Red Elastic Collar — an integral part of the nut — grips bolt threads, because its inside diameter is smaller than bolt diameter. Permits accurate bolt loading — maintains accurate adjustments.
APPROVED SELF-LOCKING FASTENERS
FOR ARMY AND NAVY AIRCRAFT, ORDNANCE AND SIGNAL CORPS EQUIPMENT
Free AN-ESNA Conversion Chart
ESNA
ELASTIC STOP NUTS

neering and scientific societies, colleges, universities and similar educational institutions.

♦

The American Lava Corp., Chattanooga, Tenn., has appointed **John A. Green Co.,** Dallas, Tex., as sales representative for the southwestern states of Texas, Oklahoma and Arkansas. The services of Bruce M. Williams and John A. Green will be available for technical problems involved in application of AlSiMag custom-made technical ceramics.

♦

Alfred W. Russell has announced the formation of **Russell Reinforced Plastics Corp.** to manufacture low pressure laminates and flat board stock of Fiberglas-polyester construction. With main offices in Hicksville and plants in Lindenhurst, L. I., N. Y., the company is already in production, operating 36 hydraulic and compressed air type presses.

♦

Onsrud Machine Works Inc., Chicago, manufacturer of high-speed production wood and nonferrous metalworking machinery and equipment, has appointed **Babcock Machinery Co.,** 1904 Times Bldg., New York, N. Y., as its representative in the New York area.

♦

So that potential users can see blast cleaning and dust control equipment in operation, the **Pangborn Corp.** has opened a new demonstration room at its plant in Hagerstown, Md. Users of blast cleaning and dust control equipment can send or bring samples of work to Hagerstown and have it processed by Pangborn machines.

♦

Peerless Engineering Sales Ltd., Toronto, Canadian representative for **Reeves Pulley Co.,** Columbus, Ind., has opened a branch office and warehouse at 1231 Ste. Catherine St. W., Montreal, Quebec. A complete line of Reeves units and replacement parts will be carried in stock. Len Hackert, a member of the Toronto sales staff for many years, is manager of the new office.

♦

A new laboratory which doubles the capacity of the company's development and research divisions has been announced by **Tousey Varnish Co.,** Chicago. The company can now offer its customers and prospective customers a complete service of analysis and recommendation, as well as immediate action on sample requirements.


SELL YOUR BRASS MILL SCRAP PROMPTLY
To keep production rolling
Help lick the shortage of brass and copper. Make sure that every pile of brass mill scrap in your factory – even if it is only a hundred pounds – is sold at once.
Not only is this scrap immediately salable on today's market – it will bring vital metal to the production lines where it is sorely needed.
Both the defense program and civilian needs require more and more copper and brass. To prevent increased shortages and even more stringent regulations, see that *your* brass mill scrap is moved promptly.
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WATERBURY 20, CONNECTICUT • SUBSIDIARY OF KENNECOTT COPPER CORPORATION
• The Nation's Headquarters for Brass & Copper
Albany† Atlanta Baltimore Boston Chicago Cincinnati
Cleveland Dallas Denver† Detroit Houston† Indianapolis
Kansas City, Mo. Los Angeles Milwaukee Minneapolis Newark New Orleans
New York Philadelphia Pittsburgh Providence Rochester† St. Louis
San Francisco Seattle Waterbury
(†sales office only)

Meetings

AND EXPOSITIONS

Jan. 14-17—
American Management Association. General management conference to be held at the Biltmore Hotel, Los Angeles, Calif. Additional information may be obtained from society headquarters, 330 West 42nd St., New York, N. Y.

Jan. 21-25—
American Institute of Electrical Engineers. Winter general meeting to be held at Hotel Statler, New York, N. Y. H. H. Henline, 33 West 39th St., New York 18, N. Y., is secretary.

Jan. 31-Feb. 1—
American Society for Metals. First midwinter technical meeting to be held at the William Penn Hotel, Pittsburgh, Pa. W. H. Eisenman, 7301 Euclid Ave., Cleveland 3, O., is national secretary.

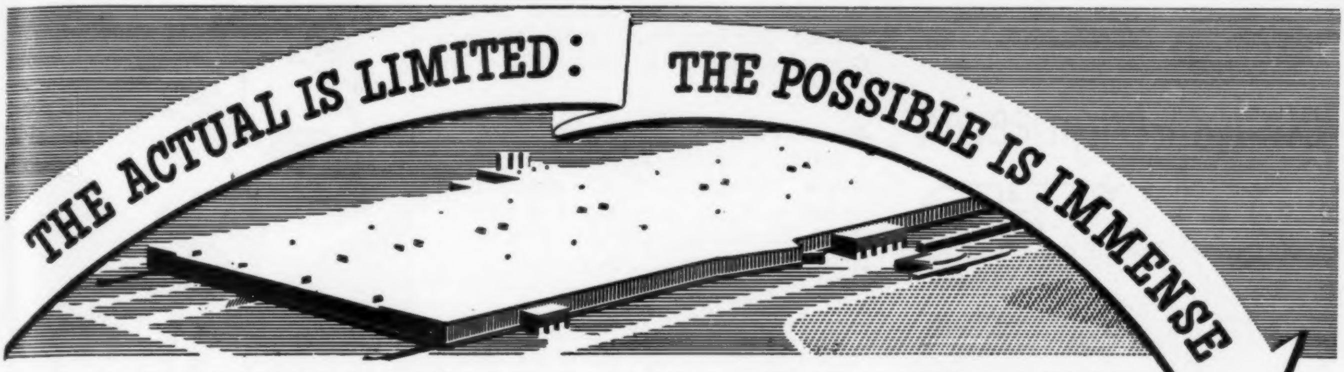
Feb. 7-8—
Instrument Society of America. Power plant symposium to be held at the Hotel Statler, New York, N. Y. Richard Rimbach, 551 5th Ave., New York 17, N. Y. is executive secretary.

Jan. 14-17—
Plant Maintenance Conference. Third conference to be held concurrently with the Plant Maintenance Show at Convention Hall, Philadelphia, Pa. Additional information may be obtained from the exposition management, Clapp & Poliak Inc., 341 Madison Ave., New York 17, N. Y.

Jan. 14-18—
Society of Automotive Engineers. Annual meeting to be held at Hotel Book-Cadillac, Detroit, Mich. John A. C. Warner, 29 West 39th St., New York 18, N. Y., is secretary and general manager.

Jan. 16-18—
Society of Plastics Engineers. Eighth annual national technical conference sponsored by the Chicago section of the SPE to be held at the Edgewater Beach Hotel, Chicago, Ill. Mrs. Bess R. Day, 409 Security Bank Bldg., Athens, O., is executive secretary.

Jan. 18—
Malleable Founders' Society. Semiannual meeting to be held at Hotel Cleveland, Cleveland, O. Additional



NEW LINCOLN PLANT CREATED BY INCENTIVE-INSPIRED CO-ACTION IN DEVELOPING POSSIBILITIES IN PRODUCT

BUILDS LIGHTER, STRONGER PRODUCTS AT HALF THE COST

By **Robert F. Christian,**
General Manager, J. D. Christian Engineers
San Francisco, California

ORIGINALLY, our gearmotor housings were made from gray iron. However, during World War II we developed many techniques in welding these gear cases that are saving materials, manpower and shop cost. Among the advantages gained by our conversion to welded steel are the following benefits:

1. Savings in weight alone have proved important to equipment builders who use our products and who must ship their machinery "prepaid" to all parts of the country. The weight savings also allow lightening up supporting structures to save construction costs of our customers' products.

2. Shorter shop schedules are improving our own deliveries by eliminating the demand for core work formerly required on our original designs and which was not attractive to suppliers.

3. With welded construction, our products are more adaptable for use on special machines. For example, the relocating of mounting bolt holes is now easier.

4. Ninety per cent of the machining time formerly required has been eliminated.

Conservatively speaking, our shop turns out comparable equipment for about half of the original cost in gray iron.

Similar savings are undoubtedly possible on many of your present and proposed products. A Lincoln Welding Engineer will gladly work with your designers to show how you can benefit with welded steel. Call or write.

PROPER DESIGN IN WELDED STEEL ALWAYS IMPROVES PRODUCT AND LOWERS COSTS

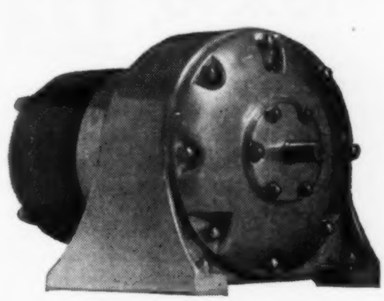


Fig. 1. Original Construction of gear motor housing. *Required 90% more machining.* Weighed 175% more than welded steel.

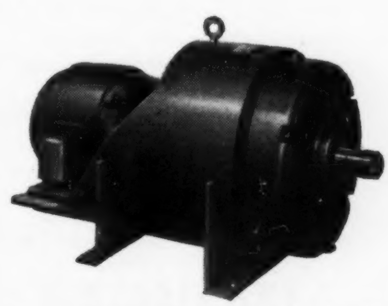


Fig. 2. Present Weldesign in Steel. *Costs 50% less.* Further saves shipping costs "prepaid" by manufacturers, using the product.

HERE'S PROOF

Machine Design Sheets are available to designers and engineers. Simply write on your letterhead to Dept. 11,

THE LINCOLN ELECTRIC COMPANY

CLEVELAND 17, OHIO

information may be obtained from society headquarters, 1800 Union Commerce Bldg., Cleveland, O.

Jan. 28-Feb. 1—
Institute of the Aeronautical Sciences. Twentieth annual meeting to be held at the Astor Hotel, New York, N. Y. R. R. Dexter, 2 East 64th St., New York 21, N. Y., is secretary.

Mar. 3-7—
American Society for Testing Materials. Spring meeting to be held at the Statler Hotel, Cleveland, O. Additional information may be obtained from society headquarters, 1916 Race St., Philadelphia 3, Pa.

Mar. 4-6—
Society of Automotive Engineers. Passenger car, body and materials meeting to be held at the Book-Cadillac Hotel, Detroit, Mich. John A. C. Warner, 29 West 39th St., New York 18, N. Y., is secretary and general manager.

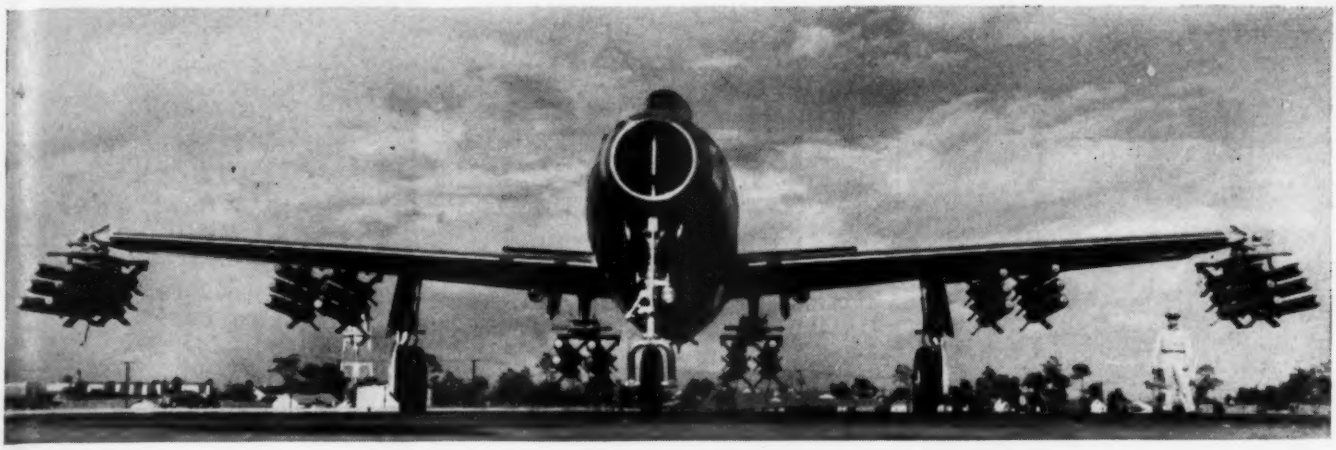
Mar. 7—
Malleable Founders' Society. Eastern sectional meeting to be held at the Commodore Hotel, New York, N. Y. Additional information may be obtained from society headquarters, 1800 Union Commerce Bldg., Cleveland, O.

Mar. 11-14—
Society of the Plastics Industry. Fifth national plastics exposition to be held at Convention Hall, Philadelphia, Pa. William T. Cruse, 67 West 44th St., New York 18, N. Y., is executive vice president.

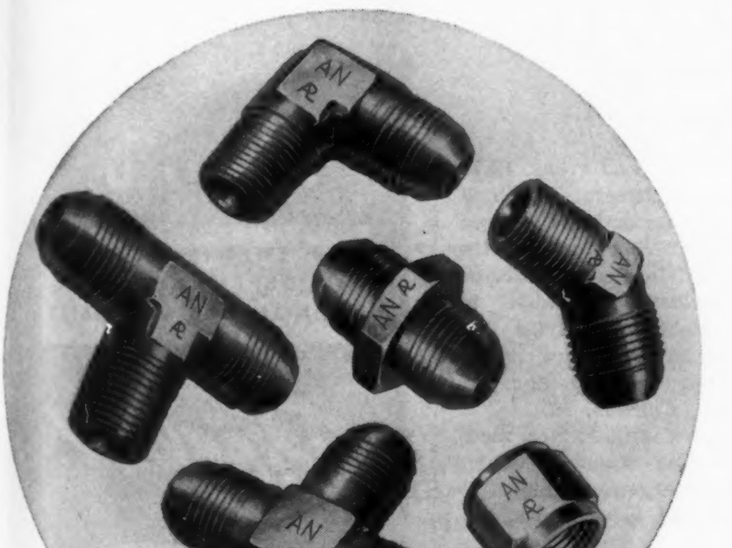
Mar. 17-19—
Midwestern Conference on Fluid Mechanics. Second conference to be held at Ohio State University, Columbus, O. Additional information may be obtained from Professor Geoffrey Keller, conference secretary.

Mar. 17-21—
American Society for Tool Engineers. Twentieth annual meeting and industrial exposition to be held at the International Amphitheatre in Chicago, Ill. Harry E. Conrad, 10700 Puritan Ave., Detroit 26, Mich., is executive secretary.

Mar. 18-19—
Steel Founders' Society. Annual meeting to be held at the Edgewater Beach Hotel, Chicago, Ill. Additional information may be obtained from society headquarters, 920 Midland Bldg., Cleveland, O.



Free-Machining ENDURO Flies with the Thunderjets



Parker AN fittings, manufactured of Type 316 ENDURO Stainless Steel by Parker Appliance Company, Cleveland, Ohio, fly with Republic Thunderjets and with many other aircraft.

Machined fittings for aircraft hydraulic control systems must, above all, be strong and tough. They must contain pressures as high as 3000 psi and possess high safety margins. They must fight off all attacks of corrosion and fatigue. They must resist the effects of sudden changes in temperature and atmospheric pressure.

The material to provide all these qualities, yet be economical to forge and to machine? It's ENDURO Stainless Steel, available in free-machining grades, either hot rolled or cold drawn bars.

ENDURO responds uniformly to forging—and, as these fittings indicate, is readily machinable.

ENDURO Cold Finished Bars are processed by Republic's Union Drawn Steel Division to provide close tolerances, accuracy of section, uniform soundness and fine surface finish. Two grades are approximately 90% as machinable as Bessemer screw stock.

To help you use ENDURO to cut costs and speed production of stainless steel bar parts, Republic offers you prompt, competent metallurgical assistance. Just write:

REPUBLIC STEEL CORPORATION
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Export Department: Chrysler Building, New York 17, N.Y.

Republic ENDURO® FREE-MACHINING STAINLESS STEEL

REPUBLIC STEEL ®

Other Republic Products include Carbon and Alloy Steels—Pipe, Sheets, Strip, Plates, Bars, Wire, Pig Iron, Bolts and Nuts, Tubing

New Machines

Business Equipment

BILLING MACHINE: All-electric model features ten-key keyset decimal tabulation. Tabulator enables rapid positioning of carriage to predetermined decimal location. Designed for the preparation of forms requiring more than one copy. Uses continuous forms. Has ten pressure-control settings for desired number of carbon copies. Includes electric intermediate and full carriage return control, palm tabulator, adjustable form-measuring gage, pivoting transparent form-cutting knife, choice of floating sheet or roll carbon paper. *Underwood Corp., New York, N. Y.*

Domestic

ELECTRIC RANGES: Two models with lighted pushbutton controls for the four surface units and ovens. Have clock type oven timers; matching automatic time measures. Equipped with full-length cooking surface lamp mounted in metal reflector on backsplasher. Available in single and double-oven styles. Ovens equipped with interior lights, aluminum broiler pans, metal racks. *Hotpoint Inc., Chicago, Ill.*

AUTOMATIC DISHWASHERS: New models hold largest standard dinner plates; have larger container to accommodate 60 pieces of silverware, 62 pieces of china. New center baffle improves washing action. Revised water inlet system meets all plumbing and sanitary codes. Features front opening, overhead spraying, new washing action that prevents glasses from spotting. *Hotpoint Inc., Chicago, Ill.*

Heating and Ventilating

UNIT HEATER: Gas-fired model for natural, mixed, manufactured and propane gas. Features maximum heat transfer through use of internal baffles and individual burners for each heat exchanger tube. Five models available with heat input of 55,000 to 200,000 Btu. *United States Radiator Corp., Detroit, Mich.*

Metalworking

HAND DRILL: Rated for heavy-duty, continuous production service. Size,

¼-in.; features ball bearings, heat treated gears, splined gear shafts and aluminum housing. End handle unit available for standard speed or low speed. Standard-speed unit has no-load speed of 1700 rpm (optional 2500, 3500 or 5000 rpm); low-speed unit has no-load rating of 600 rpm (1000 rpm optional). Powered with Black & Decker universal motor; available for 115 or 220-v power lines. Weight of drill, 3¼ lb. *The Black & Decker Mfg. Co., Towson, Md.*

POWER PRESS BRAKES: Expanded line includes sizes from 120 to 1000 tons capacity for forming mild steel from 3/16 to 1 in. thick, 6 to 20 ft long. New wedge type ram pressure release on all models releases ram in event of bottoming of dies. Models of 350 tons capacity and larger can be furnished with two-speed transmission affording approximately 4 to 1 speed ratio. Special throat depth, stroke and shunt height available. *Columbia Machinery and Engineering Corp., Hamilton, O.*

HEAT SEALER: Model UHS-54 makes seals ranging from ¼ to 1¼ in. wide and up to 54 in. long. Designed for all types of heavy coated or laminated heat-sealing barriers. Heating elements maintain temperatures up to 550 deg on sealing surface. Temperature is controlled by a thermostat and is adjustable to suit type of material being sealed. Uses up to ¾-ton pressure. Controlled and operated by foot-control switch. *Automatic Scale Co. Inc., New York, N. Y.*

HYDRAULIC COPYING LATHE: High-speed unit built by H. Ernault-Batignolles Machine Tool Div. of Batignolles-Chatillon Locomotive Co., France. Designed for carbide and diamond tools, and freedom from vibration even at top spindle speed of 3600 rpm. Copies to within ±0.0004-in. on diameter. Rear tool maintains ±0.0002-in. on diameter, and axial lengths are copied to within 0.0015-in. Copy turns from flat template or turned master. Setting time for different components, 10 to 20 minutes. Can also be used, without adjustments, as ordinary high-speed center lathe. *H. E. B. Machine Tools Inc., New York, N. Y.*

PRESSES: Capacities from 125 to 400 tons. Blank, draw, form and pierce sheet metal parts. Illuminated beds eliminate die breakage, fluorescent tubes at each end of press bed providing light to give operator full vision at all times. Bed drilled for die cushion installation, ram length equal to

bed length, no gib interference, and large gaps in columns for feeding of coil stock from sides. *Thunder Bay Manufacturing Corp., Detroit, Mich.*

TRANSFER MACHINE: Machines flywheel housings completely except for disk grinding three faces. Includes 13 stations—one each for loading, milling, rough boring, finish boring, and unloading; six for drilling and chamfering; two for tapping. Features palletized workholding fixtures which hold parts during all operations and during movement between stations. Fixtures are returned to loading station automatically by integral conveyor. Cleaning unit automatically removes chips from fixtures; vibrating type conveyor carries chips away. Equipped with Cross Toolometer which automatically stops machine when tools need resharpening. *The Cross Co., Detroit, Mich.*

INDEXING TABLE: Designed to provide high accuracy in use with milling machines, jig borers, die sinkers, etc., for indexing and rotary milling. May be used in either vertical or horizontal position. Engraved glass degree scale in conjunction with circular dial makes possible readings accurate to 0.0005-in. at periphery of 12-in. circle. Turntable is supported at its periphery about entire circumference. Forged spindle mounted on preloaded taper roller-bearing. Antifriction bearings used for all rotating shafts. Clutch permits hand rotation of turntable. *F. T. Griswold Mfg. Co., Wayne, Pa.*

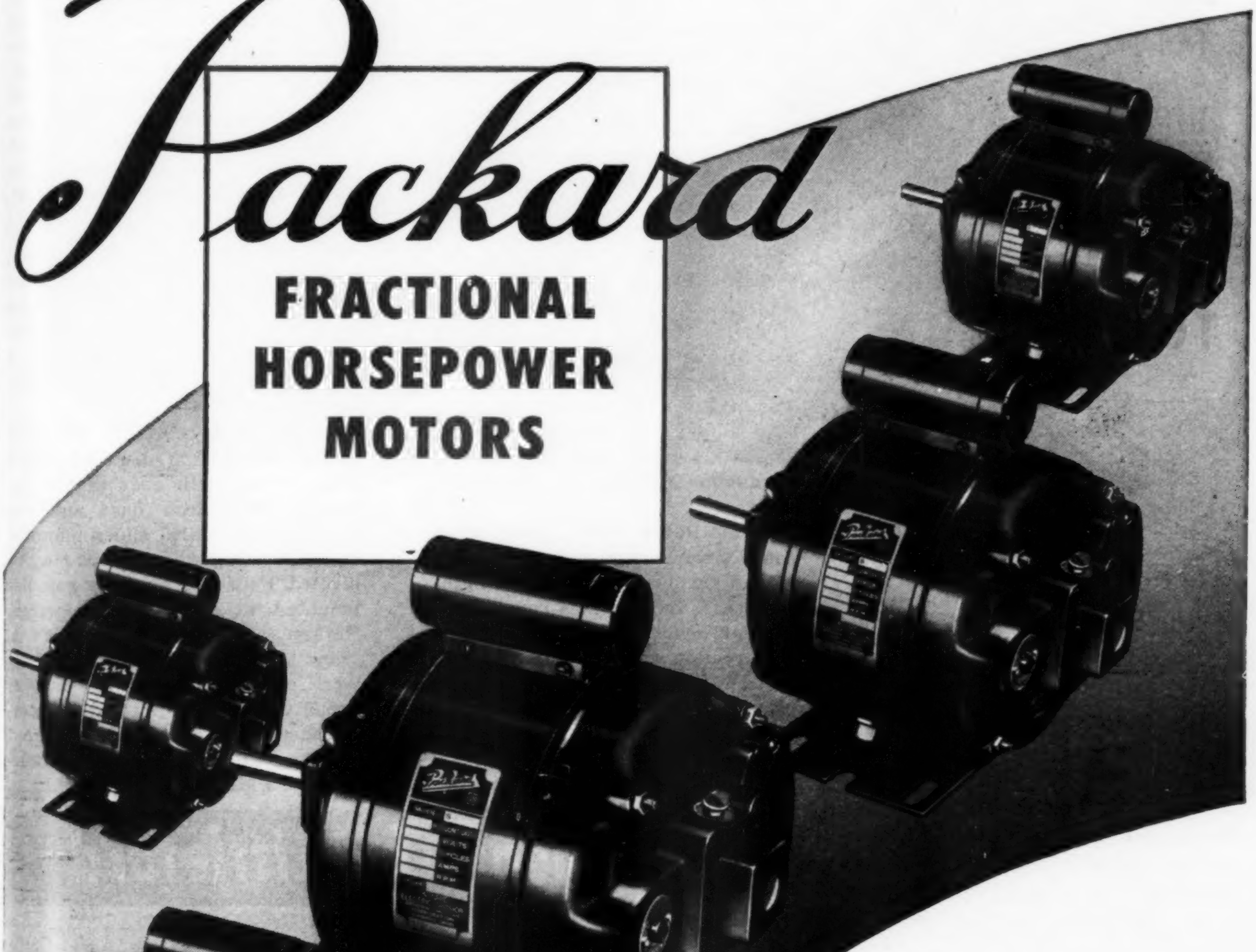
WELDING PRESS: Gap-frame unit for use with welding fixtures in mass production of spot-welded assemblies. Permits conveyorized feeding from side to side, providing accessibility of work from either front or back of press. Slide located in lower portion of press; work stroke is upward. Pneumatic friction clutch, flywheel and motor located at top of press. Operating cycle divided into thirds—for upstroke, dwell, downstroke. Machine can be started and stopped at any point in cycle under load. Slide widths range from 42 to 84 in. with 12-in. depth on all widths. Larger or smaller dimensions built to order. *E. W. Bliss Co., Canton, O.*

HAND SCREW MACHINE: Designed to cover field of former "Wire Feed" screw machines. Include full antifriction bearing spindles with positive chain drive at all speeds and completely enclosed drive. Almost 200 two-speed combinations provided with ratios of high to low speeds from 1.6:1 to 13:1. Has one-lever spindle control and mush-

Known Everywhere for Superior Performance

Packard

FRACTIONAL HORSEPOWER MOTORS



Your products gain prestige when they are powered with Packard fractional horsepower motors. The Packard motor's name is so widely known—so greatly respected—that customers value your product more when they see it is Packard powered.

For more than thirty-six years, Packard has specialized in the manufacture of fractional horsepower motors . . . and built an outstanding reputation for uniformity of quality in motor products. Today, with greatly increased facilities for engineering and manufacturing, Packard can offer you more, in service and products, than ever before.

So take advantage of Packard's ability to serve you and your products now. Choose Packard fractional horsepower motors and get thoroughly dependable quality—while you win added prestige for your product.

PACKARD ELECTRIC MOTORS for Washing Machines • Ironers • Driers • Dishwashers • Disposal Units • Water Pumps • Compressors • Power-Driven Bench Tools • Stokers • Furnace Blowers • Oil Burners • Ventilators • Milk Separators • Milking Machines and many other applications.

GM GENERAL MOTORS

Packard REG. U.S. PAT. OFF. TRADE MARK

Packard Electric Division, General Motors Corporation, Warren, Ohio

DEPENDABLE APPLIANCE MOTORS FOR THIRTY-SIX YEARS

room-top trip lever for automatic chuck and bar feed. Automatic oiling provided for all major mechanisms and bearing surfaces. *Brown & Sharpe Mfg. Co., Providence, R. I.*

BLADE GRINDER: For grinding and finishing various types of bucket blades using abrasive or felt belts by offhand grinding method. Equipped with extended housings to mount platen rolls up to 4 in. wide, and ball bearing spindles for contact wheels up to 1 in. wide by 3 in. diameter. Spindles permit 180-degree wrap of abrasive or felt belt at point of contact. Contact wheels match contour of various types and forms of blades. Available in either single or duplex type for performing two operations on one machine. Uses belts up to 4 in. wide by 115 in. long running at 5000 surface feet per minute. *Production Machine Co., Greenfield, Mass.*

HYDRAULIC POWER UNIT: Designed to convert slow-speed hydraulic presses and extrusion machines to higher speeds. Incorporates conventional high-low pressure and volume construction, with pressure-compensated high pressure circuit to hold pressure cycle at exact speeds desired. Includes electric controlled valving, decompression unit, micronic filters, large oil reservoir tank. Sizes up to 125 hp at 500 psi. *The Rucker Co., Oakland, Calif.*

DRILLING MACHINE: For opposed drilling and counterboring operations in sequence. Drills No. 44 hole and No. 30 counterbore with a combination tool; then tool from opposite side reams part way a No. 38 hole 0.004-in. off the center line of the first operation. Machine can be adapted to other drilling operations. *Govro-Nelson Co., Detroit, Mich.*

INDEXING TABLE: For light milling, drilling, tapping, etc. Mounted on large ball bearings with large hardened steel stops. Air operated; all moving parts enclosed. Size, 9 in diameter. *H. G. Weber & Co. Inc., Kiel, Wis.*

DOUBLE SPINDLE GRINDER: For finishing antifriction bearing roller ends. Grinds both ends in one operation. Cast iron base supports heads on dovetailed slides mounted on ball bearing ways. Heads may be pivoted. Four-inch grinding spindles mounted in antifriction bearings. Rotary attachment mounted on front of machine drives rotary work carrier at a variable rate of speed within a 3 to 1 ratio. Interchangeable rotary carriers permit grinding of various sizes of bear-

ing rollers. Parts are loaded manually, unloaded automatically. *Gardner Machine Co., Beloit, Wis.*

WELDING MACHINE: Heavy-duty unit designed for all industrial uses requiring d-c welding. Stack failure prevented by high-velocity downdraft of cool air directed over rectifier stacks before being passed through other parts of machine. Primary coils are raised and lowered on ball bearing jacks. Available in 200, 300 and 400-amp ratings. *A. O. Smith Corp., Milwaukee, Wis.*

PORTABLE CUTTING TOOL: Combination milling machine, planer and router. Cuts and shapes wood, plywood, plastic, masonite, stone, paper, fabric and nonferrous metals and alloys. Follows concave and convex curves and at the same time can be moved upward or downward to cut in two directions. Furnished with one cutting head and two high-speed adjustable cutting blades. Has a 115-v, ac-dc 18,000-rpm motor. *Composite Die Supply Co., Detroit, Mich.*

FINISHING MACHINE: For drilling and chamfering cylinder heads. Drills 12 angular holes and countersinks 8 manifold mounting holes in 170 pieces per hour at 100 per cent efficiency. Features heavy-duty index table which incorporates a manual as well as an automatic control cycle. Fluid motor drive prevents damage to table due to jamming. *The Cross Co., Detroit, Mich.*

Materials Handling

LIFT TRUCK: Improved Model 20 has new upright assemblies and load arm pins which permit the use of optional additional counterwight on standard 2000-lb capacity model, thus increasing or decreasing load-center or load capacity as needed. Available in standard 2000-lb capacity with 24-in. load center, lightweight model, and several ratings with 15 or 24 in. load centers. *Hyster Co., Portland, Ore.*

CONVERGING SECTION: For roller conveyors. Available for all widths from 6 to 36 in. Handles all types of packages, boxes, cartons and tote pans and components in process. *Sage Equipment Co., Buffalo, N. Y.*

CLAMPING ATTACHMENT: Hydraulically-operated clamping forks can be fitted with self-aligning rubber-faced gripping pads to handle drums and other cylindrical objects. Horizontal movement of forks produced by two double-acting hydraulic rams synchronized to give uniform motion to each fork. Min-

imum distance between forks, 12 in.; maximum distance, 27 in. Attachment can be used on 1000, 1500 and 2000-lb Spacemaster "49" models. For use in close quarters. *Lewis-Shepard Products Inc., Watertown, Mass.*

SHEET HANDLING TRUCK: Has seven openings each 4⅜ in. wide, 53 in. high. Deck portion of truck is open and fitted with short rollers on ball bearings to allow easy entrance and removal of sheets. Floor locks on each side of truck keep it stationary and in position for loading or unloading. Size: 3 ft wide, 8 ft long, 6 ft high; capacity, 6000 lb., *Market Forge Co., Everett, Mass.*

MACHINE TENDER: Handles variety of parts and small items. All welded construction. Size: 30 in. long, 16 in. wide, 32 in. high. Top deck, 2 in. deep; lower deck, 3 in. deep. Both have turned edges all around. Has four 5 by 1 in. plain bearing wheels, swivel at handle or rear end; rigid in front. Can be equipped with rubber-tired wheels. *Palmer-Shile Co., Detroit, Mich.*

Plant Equipment

LOW-TEMPERATURE CABINET: Provides temperatures as low as −40 F. Used for rivet cooling, shrink fit assembly, size-stabilization in metal, storing punched and formed aluminum alloy parts, applying subzero compressed air to metal cutting tools and treating hardened steels. Foot treadle opens lid. Powered by 1/3-hp, 110-v a-c hermetic unit. Size: 40 by 36 by 32 in.; capacity, 2½ cu ft. *Brewer-Titchener Corp., Binghamton, N. Y.*

AIR COMPRESSOR: Four-cylinder, two-stage unit for 3 and 5-hp air compressing outfits. All sizes available for either 125 or 175-psi service. Generated heat is dissipated by finned cylinder heads, blocks, intercoolers and check valve. *DeVilbiss Co., Toledo, O.*

PORTABLE ELECTRIC OVEN: Has eight drawers that permit baking of different materials simultaneously or materials inserted at intervals. High-temperature magnesium strip heaters provide heating. Temperature controlled by thermostat; adjustable up to 325 F. Operates on 110-v current. Forced circulation by fan driven by electric motor; adjustable damper controls volume of air circulation. Can be used in group or stacked one on top of the other. Outside dimensions: 30 by 25 by 24 in.; inside dimensions 28½ by 24 by 20½ in. Drawers are 2 by 11½ by 23 in. *Grieve-Hendry Co. Inc., Chicago, Ill.*